VAN NOSTRAND'S SCIENCE SERIES.

No. 17.—WATER AND WATER SUPPLY. By PROF. W. H. CORFIELD, M. A., of the University College, London.

No. 18.—SEWERAGE AND SEWAGE UTILIZATION. By PROF. W. H. CORFIELD, M. A., of the University College, London.

No. 19.—STRENGTH OF BEAMS UNDER TRANSVERSE LOADS. By PROF. W. ALLEN, Author of "Theory of Arches." With Illustrations.

No. 20.—BRIDGE AND TUNNEL CENTRES. By JOHN B. MCMASTERS, C. E. With Illustrations.

No. 21.—SAFETY VALVES. By RICHARD H. BUEL, C. E. With Illustrations.

No. 22.—HIGH MASONRY DAMS. By JOHN B. MCMASTERS, C. E. With Illustrations.

No. 23.—THE FATIGUE OF METALS UNDER REPEATED STRAINS, with various Tables of Results of Experiments. From the German of PROF. LUDWIG SPANGENBERG. With a Preface by S. H. SHREVE, A. M. With Illustrations.

No. 24.—A PRACTICAL TREATISE ON THE TEETH OF WHEELS, with the Theory of the Use of Robinson's Odontograph. By S. W. ROBINSON, Prof. of Mechanical Engineering. Illinois Industrial University.

No. 25.—THEORY AND CALCULATIONS OF CONTINUOUS BRIDGES. By MANSFIELD MERRIMAN, C. E. With Illustrations.

No. 26.—PRACTICAL TREATISE ON THE PROPERTIES OF CONTINUOUS BRIDGES. By CHARLES BENDER, C. E.

No. 27.—ON BOILER INCRUSTATION AND CORROSION. By F. J. ROWAN.

THE STEAM-ENGINE INDICATOR AND ITS USE.

A Guide to Practical Working Engineers

FOR

GREATER ECONOMY AND THE BETTER WORKING OF STEAM-ENGINES.

THE INDICATOR, AND WHAT ITS OBJECT IS; ITS CONSTRUCTION AND ACTION THOROUGHLY EXPLAINED; TOGETHER WITH THE METHOD OF CALCULATING THE HORSE-POWER OF ENGINES AS SHOWN BY THE DIAGRAMS.

BY

WILLIAM BARNET LE VAN.

ILLUSTRATED WITH TWENTY ENGRAVINGS.

REPRINTED FROM "THE MECHANICAL ENGINEER," WITH MUCH ADDITIONAL MATTER.

NEW YORK:

D. VAN NOSTRAND, PUBLISHER,

23 MURRAY AND 27 WARREN STREETS.

1884.

ELECTROTYPED AND PRINTED
BY RAND, AVERY, & COMPANY,
BOSTON, MASS.

PREFACE.

At the request of Egbert P. Watson, editor of "The Mechanical Engineer," the author prepared for publication in that journal a series of articles on "The Steam-Engine Indicator," which were published, and attracted the attention of readers interested or engaged in the study and the use of marine, locomotive, and stationary engines. These articles, rewritten, revised, with much additional matter, make up this treatise, which is intended as elementary.

As connected with the use and application of the indicator, an endeavor has been made to explain the most important parts of the theory and action of steam, and to show the modes of working engines that

have been found to be the most advantageous, either for the development of power, or the saving of expense. The author has taken whatever he deemed valuable or necessary for the work from the best accredited writers on these topics, and offered suggestions of his own; his purpose being to so treat the whole subject as to make it easily understood by even those least familiar with the operation of steam, and practically useful to those whose education has been, and must be, rather in the engine-room than in the class-room.

W. Barnet Le Van.

Philadelphia.

THE STEAM-ENGINE INDICATOR, AND ITS USE.

BEFORE entering upon the subject of the action of steam in the cylinder of an engine, a few preliminary observations will be necessary to guard against mistakes which sometimes arise with reference to the technical terms, and scales of measurement of pressures, temperatures, etc.

ELEMENT.

The term "element," as used, means a body which cannot, as far as we know, be divided or decomposed. It is unalterable and indestructible, and therefore called a simple body, or element.

A first or constituent principle of any thing; that which admits not of division or decomposition into two or more ingredients of unlike properties; a simple or unde-

compounded body, — as, "The *elements* of water are oxygen and hydrogen."

Formerly, and still in popular language, *earth*, *air*, *water*, and *fire* are called the four elements, because they were formerly deemed first principles.

> "The *elements* be kind to thee!"
>
> *Shakspeare.*

The proper habitation or sphere of any thing; suitable state.

> "A fish is out of his *element* when he is not in the water." — *Milton.*

Elements, however, combine with one another, and lose their individuality, so to speak, in the compound. Thus two simple bodies, gases, hydrogen and oxygen, unite in certain definite proportions, and form water; but the water can be again decomposed into its constituents.

Air is called a compound: it is a mixture of gases which preserve their individuality, as sand and sugar would do if they were mixed in a vessel.

The principal material elements required for the support of life are *oxygen*, *hydrogen*,

nitrogen, and *carbon;* and the simple physical elements are *force*, *velocity*, and *time*, which constitute the functions *light*, *heat*, *power*, *space*, and *work*.

THE PROOF OF ATMOSPHERIC PRESSURE.

The real cause of atmospheric pressure is the weight of the atmosphere. When the air is removed from any surface of a body, the pressure of the air at once manifests itself.

The removal of the air from any body produces in the space from which it is taken a condition which is called a "*vacuum.*" The literal meaning of the term *vacuum* is a condition of space unoccupied by matter, "*empty.*" The utility of this natural phenomenon was experimented on as far back as 1634, by Torricelli, a pupil of the renowned astronomer and philosopher Galileo, in the following manner: —

A glass tube about three feet long, and about one-quarter inch internal diameter, was sealed at one end, and filled with mercury. The open end being stopped by placing the thumb over it, the tube was then inverted; and the lower end, covered

by the thumb, was inserted in a vessel containing mercury. The tube still remaining vertical, the thumb being removed, the mercury sank about six inches from the sealed end, which showed about thirty inches (29.9218 exact) of mercury in the tube from the level of that in the vessel. The mass in the tube was, therefore, held by the atmosphere; or, the pressure of the latter on the mercury in the vessel prevented that in the tube descending below the six inches alluded to. Now, if the top end of the tube had been opened, the mercury in it would have sunk into the vessel, because the pressure of the atmosphere would have been in equilibrium on the inside and outside of the tube; and thus the mercury would fall, owing to gravity.

The application of this experiment for practical purposes was thus carried out: A tube *one* square inch in area, filled as before, and inverted likewise in a vessel of suitable dimensions, retained the mercury thirty inches high (29.9218) under the same circumstances. Then, as two cubic inches (2.036 exact) of mercury equals about one

pound avoirdupois weight, and there are thirty cubic inches resisting the atmosphere, the matter is resolved into a simple calculation. Thus: —

W = weight of the mercury in the tube.
C = cubic inches of mercury to equal one pound.
x = pressure of the atmosphere in pounds per square inch.

Then —

$$x = \frac{W}{C} = \frac{30}{2} = 15 \text{ pounds on the square inch.}$$

This is termed a *perfect* vacuum; or, by using the above exact fractions, it will be 14.7 pounds nearly.

STEAM.

Its density is equal to the pressure of the atmosphere, or 15 (14.72 exact) pounds per square inch; and, unless confined, the temperature of water cannot be raised above the boiling point. The addition of 1° in the temperature of steam will increase its volume 0.00202 of the volume occupied by the fluid at 32°.

One cubic inch of water, evaporated under ordinary pressure of atmosphere, is converted into 1,700 cubic inches of steam, or

nearly one cubic foot. It exerts a mechanical force equal to raising 2,120.14 pounds one foot high.

One cubic foot of steam, evaporated in one hour, is equal to one horse-power; 27,222 cubic inches of steam, at atmospheric pressure, equals one pound avoirdupois; one pound of water, heated from 32° to 212°, requires as much heat as would raise 180 pounds 1°.

One pound of water at 212°, converted into steam at 212° (pressure of 14.72 pounds), absorbs as much heat for its conversion as will raise 966.6 pounds of water 1°. Steam at a pressure of one pound per square inch will support a column of mercury (60° Fah.) 2.0376 inches.

HEAT AND WORK.

The earliest philosophers discredited the materiality of *heat*, and their theories approached very closely to those now universally accepted. These, however, were suppositions only, until Benjamin Thompson — Count Rumford — proved in 1798, experimentally, that *heat* could not be a ma-

terial substance, but was probably a manifestation of *work*. Sir Humphry Davy, a little later (1799), published the details of an experiment which conclusively confirmed these deductions from Rumford's *work*.

Bacon and Newton, and Hook and Boyle, seem to have anticipated — long before Rumford's time — all later philosophers, in admitting the probable correctness of that modern dynamical, or vibratory, theory of heat which considers it a mode of motion; but Davy, in 1812, for the first time stated plainly and precisely the real nature of heat, saying, "The immediate cause of the phenomenon of heat, then, is motion; and the laws of its communication are precisely the same as the laws of the communication of motion." The basis of this opinion was the same that had been noted by Rumford.

Dr. Mayer of Heilbronn, in 1842, suggested the identity of *heat* with *work*. James Prescott Joule of Manchester, in 1843, proved, by a long series of experiments, that the production of heat was attended by the disappearance of a definite amount of mechanical *work*.

The labors of Mayer and Joule resulted in the important discovery of the dynamical value of *heat*, or, as it is usually termed, the mechanical equivalent of heat. This was found after making a very large number of experiments, conducted in a great variety of ways: a weight of one pound falling through 772 feet would raise one pound of water through 1° Fah.; or, what is just the same thing, a weight of 772 pounds falling through one foot.

The knowledge which this law gives us is exceedingly valuable. From it we learn that, in the very best engines that can be made, we are getting only about *ten per cent of the whole power out of the coal which is in it.*

We will take a condensing engine averaging 85 horse-power, with a consumption per hour of 153 pounds of coal; and the consumption per hour per horse-power would be —

$$\tfrac{153}{85} = 1.8 \text{ pounds.}$$

The best anthracite coal contains ninety per cent of carbon. Throwing away the other constituents, we are burning ninety

per cent of 1.8 pounds of pure carbon; or,

$$1.8 \times 0.90 = 1.62 \text{ pounds.}$$

Experiments show that a pound of carbon generates, while burning to carbonic acid, 14,500 units of heat, — that is, it gives off as much heat as will raise 14,500 pounds of water 1° Fah.; and therefore 1.62 pounds will generate —

$$14{,}500 \times 1.62 = 23{,}490 \text{ units of heat.}$$

We are therefore generating, in round numbers, 24,000 units of heat, and getting in exchange one indicated horse-power.

Above we have seen that one unit of heat is equivalent to 772 pounds raised one foot high; and, therefore, 24,000 units of heat are equivalent to —

$$24{,}000 \times 772 = 18{,}696{,}000 \text{ foot-pounds.}$$

But an indicated horse-power means 33,000 pounds raised one foot high per minute, which is equivalent to 33,000 multiplied by 60 minutes, —

$$33{,}000 \times 60 = 1{,}980{,}000 \text{ foot-pounds per hour.}$$

From this we see that we are burning coal

sufficient to raise 18,696,000 foot-pounds. Therefore we are, in fact, out of one of the very best engines, getting but *one-ninth*, or about ten per cent, of the power we should do.

VAPORS.

A vapor is a gas at a temperature near to that at which condensation occurs. All bodies assume the gaseous condition at suitable temperatures. In an intensely heated furnace (that of the electric light) even carbon has been made to appear as a gas, although only in a small quantity. Most solids liquefy before becoming gaseous; but some appear to become gases at once, and are said to sublime.

According to Professor James Thomson, this always occurs when the boiling point of the substance at the given pressure is lower than the freezing point for the same pressure.

Vapors are formed more readily *in vacuo* than in air; but, for any given temperature, the quantity of vapor which will form in a space from an exposed liquid is the same,

whether air or other gases be present or not, — the vapor being formed almost instantaneously in the second case, and requiring more or less time for formation in the first.

The pressure which this vapor eventually adds to the pressure of gases already existing in the space depends on the temperature only, and is the same, no matter what may have been the previously existing pressure. When no more liquid will change into vapor, we may say that the space is *saturated.*

A space unsaturated with vapor may be contracted, the pressure becoming greater until the maximum pressure for that temperature is reached. After this, contraction at constant temperature causes some of the vapor to be condensed to the liquid state, as the pressure has reached its maximum, and the space has become saturated.

Unsaturated vapors follow approximately the laws of gases in expanding with heat. Steam, when passing along hot pipes to the engine, may be *superheated;* and its coefficient of expansion will be found to differ very little from that of common air. By superheating steam we increase its volume,

whilst its pressure is unchanged. We also render it less liable to condense in the cylinder; and we convert into steam many particles of water which are often carried over from the foam in the boiler.

LOW PRESSURE AND HIGH PRESSURE.

Steam, or the vapor of water, when produced at the usual pressure of the atmosphere, is commonly denominated *low-pressure;* in opposition to that which is formed at a higher pressure than that of the air, and accordingly named *high-pressure* steam. In common language, however, the term "low-pressure steam" is applied to the steam which has even a pressure of several pounds on the square inch, and is therefore formed at a temperature higher than 212°. The steam-engines supplied with a condenser, when first made, used low-pressure steam, and by condensing the exhaust gained the additional pressure nearly due to the atmosphere; and were usually called *low-pressure engines*, instead of condensing engines, their proper name. In the present advanced state of

engineering, high-pressure steam is now most generally used for supplying condensing engines. They are called *condensing* and *non-condensing engines.*

ABSOLUTE PRESSURE.

It is customary to express the elastic force of steam in three ways; namely, in pounds of pressure that it exerts on the square inch, in the height of the column of mercury which it sustains, and in atmospheres. (The actual pressure of the atmosphere is continually varying, the barometric column fluctuating generally between 28.5 and 30.5 inches in height; these points in either direction being, however, but rarely reached, and still more rarely passed.) This is simple and convenient for most purposes relating to the boiler, but leads to misconception when applied to steam in the cylinder of an engine. Water evaporated in the open air is said, according to this notation, to be transformed into steam of zero pressure, instead of steam of 14.7 pounds pressure, per square inch; which pressure counterbalances that of the

atmosphere. If such steam is used in a condensing engine, the effect is said to be due to *vacuum*, which is still regarded by some people as a separate force unconnected with steam, and in fact operating on the other side of the piston. When steam of higher pressure is used, it is customary, in finding the horse-power, to add the *vacuum* to the *steam pressure*,—so carrying out the same idea.

The absolute pressure of steam is measured from zero, or perfect vacuum, and consists of the pressure as shown by the steam-gage (which only shows the pressure above atmosphere); and, as before stated, the pressure of the atmosphere is indicated by the barometer. The latter may, for all practical purposes, be taken at 15 pounds, corresponding to 30.6 inches of mercury. The vacuum gages in general use are usually graduated to agree with the scale of barometer, and the vacuum is usually stated in inches of mercury. To the steam pressure shown by gage, add 15 pounds for total pressure. Thus, if the pressure gage indicates 75 pounds, the total, or *absolute*, pressure is 90 pounds.

When the piston moves forward in an engine, the total pressure on steam side at any point in the stroke of piston is the pressure above the atmosphere, plus 15 pounds; and the total pressure for whole stroke is the mean pressure above the atmosphere, plus 15 pounds. Thus, if the mean pressure for whole stroke is 25 pounds, the total mean pressure is 40 pounds; and this 40 pounds, whether the engine is operated condensing or "low-pressure," or non-condensing or "high-pressure," is the variable factor in estimating the load on the engine.

Now, if the engine be operated non-condensing, the 15 pounds (pressure of atmosphere) on steam side is balanced by a like pressure of atmosphere on exhaust side of piston; and its effect is annihilated, or reduced to nothing. But, if the engine be operated condensing, a large proportion of the pressure of atmosphere on exhaust side of piston is removed, and an equivalent portion of the pressure of atmosphere on steam side of piston made to do useful work. With well-proportioned condensing apparatus, the pressure of atmosphere on

exhaust side of piston can be reduced nearly ninety per cent: in other words, a vacuum in the exhaust end of cylinder of (13 pounds) 26.5 inches may be maintained; and this 26.5 inches, or 13 pounds per square inch of piston, is an absolute gain, and should in all cases be utilized.

PROPERTIES OF STEAM.

Total pressure, or absolute pressure, means the steam pressure in pounds per square inch, including the pressure of the atmosphere, and is generally denoted by P; and p is used to denote the steam pressure above atmosphere, as is shown on the ordinary spring-gage. If a mercury column is used, it is shown in inches and fractions of inch. The specific gravity of mercury at 32° Fah. is 13.5959, compared with water of maximum density at 39°. One cubic inch of mercury weighs 0.49086 pounds; of which a column of 29.9218 inches is a mean balance of the atmosphere, or 14.68757 pounds per square inch, very nearly.

It is common to say that an atmosphere

is 15 pounds on the square inch, or 30 inches of mercury. This is not correct; the pressure of 15 pounds on the square inch being equal to that of a column of mercury 30.55 inches in height. One pound pressure on the square inch is equal to 2.037 inches of mercury.

The French use a column of mercury 760 millimetres in height, at the temperature of 0° C., or 32° Fah., which is as nearly as possible the mean atmospheric pressure.

EXPANSIVE POWER OF STEAM.

When a volume of air is compressed into a smaller volume, a certain amount of power is expended in compressing it, which power, as in the case of a bent spring, is given back when the pressure is withdrawn. If, however, compressed air is suddenly released into the atmosphere, the power expended in compressing it is lost. If this power can be utilized, it will be clear gain. This gain can be effected if, instead of releasing the compressed air, it is permitted to expand to its original volume. Now, the steam used to propel engines is in the con-

dition of air already compressed; and, to save the power which would be lost if the steam were suddenly released into the atmosphere, it must be used expansively; and, to use it expansively, it must be cut off,—that is, the steam port must be closed before the piston has completed its stroke. If the flow of steam to an engine be cut off when the piston has performed half-stroke, leaving the stroke to be completed by the expanding steam, it has been found by experiment that the efficacy of a given quantity of steam will be increased 1.7 times beyond what it would have been if the steam at half-stroke had been released into the atmosphere, instead of allowed to expand in the cylinder. If cut off at one-third of the stroke, the efficacy will be increased 2.1 times; at one-fourth stroke, 2.4 times; at one-fifth, 2.6 times; at one-sixth, 2.8 times; at one-seventh, 3 times; and at one-eighth, 3.2 times.

LATENT HEAT OF STEAM.

In generating steam from water, there is absorbed about five and a half times as

much heat as is required, under atmospheric pressure, to raise the temperature of the water from freezing point, 32° Fah., to boiling point, 212° Fah., — an amount of heat which, if the water were a fixed solid, would render it *red-hot* by daylight. Tested by a thermometer, the steam will show only 212° of sensible heat. It has been found, however, by experiment, that, when the steam from a pound of water at 212° is returned by condensation to water, sufficient heat is set free to raise the temperature of 5½ pounds of water from 32° to 212°: thus proving that the steam has absorbed five and a half times as much heat, in becoming steam, as the water from which it is produced exhibited in passing from 32° to 212°; and that, in the process of wholly evaporating into steam any given quantity of water, 1,000° of heat are absorbed. But, as the thermometer indicates only 212° of heat in steam, it follows that this prodigious excess of heat — the difference between 212° and 1,000° — must be stored up in the steam in some hidden, unaccountable way and condition, and is called the *latent* heat of steam.

TO WORK STEAM EXPANSIVELY.

If we take an upright cylinder one inch in diameter and at least 1,700 inches in height, pour into it one cubic inch of water; fit into it a steam-tight piston, resting on the water, so counterbalanced as to be weightless, and so arranged as to work without friction; and then place a lamp under the cylinder, — we then notice, that, so soon as the water reaches the temperature of 212°, it will begin to boil, and produce steam, and the steam will begin to push up the piston. So long as the lamp continues to burn, and the water continues to boil, so long will the steam continue to push up the piston, until all of the water has been evaporated into steam. When all of the water has so evaporated, it will be found that from one cubic inch of water there has been produced 1,700 cubic inches of steam; and as this would fill 1,700 cubic inches of the cylinder, and as the pressure of the atmosphere — the only resistance in this case to be overcome — is 15 pounds (14.7 exact) to the square inch, this experiment would show that one cubic inch of water,

wholly evaporated into steam, will push or lift, say, 15 pounds 1,700 inches, or 142 feet. If, now, the experiment be carried a little farther with a similar cylinder and piston, and 15 pounds be loaded on the piston, making, with atmospheric pressure, 30 pounds, we shall find that under this additional pressure the temperature of the water must be raised to 252°, instead of 212°, before it begins to boil, and before the steam begins to push up the piston; and that, when the whole of the water is evaporated, there will be only 850 instead of 1,700 cubic inches of steam, and the piston will be pushed or lifted up only 850 instead of 1,700 inches, or, in round numbers, 71 feet. If, then, one cubic inch of water, wholly evaporated, will produce steam enough to push or lift 15 pounds 142 feet, and 30 pounds 71 feet, it would produce steam enough to push or lift 142 times 15 pounds, or 2,130 pounds (say one ton) one foot. When, then, the steam from one cubic inch of water has pushed or lifted one ton one foot, it has done all it can do; and, if the experiment is to be re-

peated, this spent steam must be released by means of a valve, called the exhaust valve, and new steam admitted, or generated, to push or lift up the piston. The machinery used in this experiment represents simply a full-stroke, or non-expansion, engine, making one stroke; and, for each stroke made by such an engine, the utmost possible power to be got is equivalent to one ton lifted one foot for every cubic inch of water evaporated, — no more, no less. This is all the power we can get out of a steam-engine without a cut-off.

But let us experiment a little farther. Suppose we load the piston with one ton of bricks, and suppose, instead of opening the exhaust valve, we remove one of the bricks: the load being thus, to this extent, diminished, the steam, no longer compressed by the whole ton, will expand a little, and push or lift up the rest of the bricks a little farther; and, as brick after brick is removed, the steam will push or lift up the rest of the bricks farther and farther, until, the last brick having been removed, it will be found that the steam has pushed

or lifted up the piston to the full height of 1,700 inches, or 142 feet. Now, it will be seen from this experiment, that all the power which was produced by the steam as the bricks were successively removed was a clear gain, as it cost no fuel or steam other than that which had already pushed or lifted the one ton one foot, and can do no more unless and until the steam was relieved of a part or the whole of the resisting weight or pressure. This principle, the law of expanding steam, was discovered by James Watt.

RATE OF EXPANSION.

The higher the grade, or ratio, of expansion, the greater is the economy; but the result is somewhat modified by other considerations.

First, The higher the rate of expansion, the lower is the mean or average pressure throughout the stroke; and a low mean pressure involves the use of a large engine for a given power.

Second, With a high rate of expansion, the mean pressure is much lower than the

initial pressure; and, although the power of the engine is only due to the mean pressure, the strength of the engine must be sufficient to withstand the initial pressure.

Third, A very high rate of expansion also leads to a very low final pressure; and as to drive the engine itself against its own friction only, and to expel the steam from the cylinder, seldom require less than three pounds above the external pressure, it follows, that, if the steam is so far expanded that the terminal pressure falls below this, the expansion is excessive, and the reverse of advantageous.

In non-condensing engines, the lowest final pressure is determined by the pressure of the atmosphere, say 15 pounds per square inch; and 18 pounds may be taken as the lowest pressure to which steam can be expanded with advantage. If the exhaust passages are small, or the exhaust steam damp, a higher final pressure will be more economical. In condensing engines, the temperature of the condenser is generally about 100° Fah., and the pressure corre-

sponding to this is about one pound per square inch; but the presence of air in the condenser generally prevents the pressure there falling below two pounds per square inch. From four to five pounds may be taken as the lowest advantageous final pressure.

Fourth, The highest advantageous rates of expansion, even with jacketed cylinders, appear, in practice, to be between 12 and 16 times. Higher rates are, and should be, aimed at; but, with our present arrangement of engine, it is doubtful whether the increased economy of very high ratios, or grades, pays for the increased complication and the extra cost of the apparatus required to attain it. In unjacketed cylinders the limit of advantageous expansion is much under the lowest of the grades named.

In practice, the best result of steam-engines does not convert more than *ten per cent* of the heat used by it into work; and this in engines of considerable size, and with boilers and furnaces fairly efficient. In small engines it is much less; indeed, it is certain that few among the thousands of

steam-engines in daily use below five horse-power give an efficiency greater than *five* per cent. The great cause of loss is the amount of heat necessary to change the water from the liquid to the gaseous state, most of this being rejected with the exhaust either into the condenser or the atmosphere. Many attempts have been made to use liquids of lower specific heat than water, and requiring less heat for evaporation, — the principal being alcohol, ether, and carbon bi-sulphide; but, for obvious reasons, no success has been attained.

TEMPERATURE OF STEAM.

When steam is generated in a boiler, the water is heated until it arrives at the temperature of ebullition, and the elevation of temperature is sensible to the thermometer. Next, the water is converted into steam by an additional absorption of heat, which is not measured by the thermometer, and is therefore called latent. The heat is not, in fact, latent, but is appropriated in converting water into steam of the same temperature.

The pressure, as well as the density, of steam which is generated over water in a boiler, rises with the temperature; and, reciprocally, the temperature rises with the pressure and density. There is only one pressure and one density for each temperature; and thus it is that steam, produced in a boiler over water, is always generated at the maximum density and maximum pressure corresponding to its temperature. In such condition, steam is said to be saturated, being incapable of vaporizing more water into the same space, unless the temperature be raised. Saturation is therefore the normal condition of steam generated in contact with a store of water, and the same density and the same pressure are always to be found in conjunction with the same temperature.

In consequence, saturated steam over water stands both at the condensing point and at the generating point; that is, it is condensed if the temperature falls, and more water is evaporated if the temperature rises.

If saturated steam is separated from water in a space of fixed dimensions, if an

additional quantity of heat be supplied to the steam, the state of saturation ceases: the steam becomes superheated, and the temperature and the pressure are increased, whilst the density is not increased. Steam, thus surcharged with heat, approaches to the condition of a perfect gas.

TOTAL UNITS OF HEAT IN STEAM.

The total heat of steam consists of its latent heat, in addition to its sensible heat. The latent heat of saturated steam at 32° Fah., the freezing-point, was experimentally determined by Regnault to be equal to 606.5° C. (centigrade); or such that the total heat of one pound of saturated steam at 0° (zero) C. would be capable of raising the temperature of 606.5 centigrade pounds of water 1°.

At higher temperatures, the total heat of saturated steam was found to increase uniformly between the temperature 0° C. and 204.44° C. at the rate of 0.305° for each addition of temperature of one degree; and, therefore, if the temperature in degrees be multiplied by 0.305 (its specific

heat), and 606.5 be added to the product, the sum will express the total heat of saturated steam at the given temperature; or,

$$H = 606.5 + 0.305 T^{\circ},$$

where H equals the total heat of steam, and T° degrees centigrade, which, for Fahrenheit scale, will be as follows: —

$$H = \tfrac{9}{5}(606.5 + 0.305 T^{\circ});$$

or

$$H = 1091.7 + 0.305(T^{\circ} - 32^{\circ});$$

or

$$H = 1082 + 0.305 T^{\circ}.$$

In which T° being the temperature as read on Fahrenheit scale, and H being the total heat, or the heat necessary to convert one pound of water at the freezing-point into saturated steam at the temperature T°.

That is to say, that steam at 32° Fah., the freezing-point, will be equal to 1,082°; or that the total heat of one pound of saturated steam at 32° Fah. would be capable of raising the temperature of 1,082 pounds of water 1°. The water from which the steam is generated is supposed to be supplied at the temperature of 32° Fah.

The expression of the total heat repre-

sents units of heat when the weight of the steam is one pound.

If the water to be evaporated is supplied at any higher temperature than 32° Fah., the total heat to be expended in evaporating it is found by deducting the difference of temperature from the total heat.

Example. — Water is to be supplied at 60°, which is (60 − 32) 28° above 32° Fah. Then, 1,082 − 28 = 1,054.

Before Regnault made his experiments, it was supposed that the total heat of steam, or the sum of its sensible and latent heat, was the same for all temperatures; but these experiments prove that it increases with the increase of temperature in the uniform ratio of 0.305 of a degree for each degree of sensible heat: so that, as steam expands in volume, of each one degree of temperature that it loses, 0.6065 parts only become latent, or are converted into internal work; and the remaining 0.305 parts are set free, and are capable of being converted into mechanical work.

Table No. 1.

PROPERTIES OF STEAM.

Total pressure in pounds per square inch.	Temperature. Fahrenheit.	Volume water equal 1 at 40.	Units of heat from 32° to $T°$. Total per pound.	Total per cubic foot.	Latent per pound.	Latent per cubic foot.	Pressure above atmosphere.
P	$T°$	V	H	H'	L	L'	p
14.7	212.0	1740.0	1146.6	41.100	965.7	34.61	0.0
15.0	213.0	1706.0	1147.0	41.920	965.1	35.29	0.3
20.0	228.5	1288.0	1151.7	55.802	954.1	46.23	5.0
25.0	241.0	1035.0	1155.7	69.632	945.4	56.96	10.0
30.0	251.4	866.7	1158.7	83.410	937.8	67.51	15.0
35.0	260.7	745.8	1161.5	97.156	931.2	77.89	20.0
40.0	268.9	654.9	1164.0	110.87	925.3	88.14	25.0
45.0	276.2	584.1	1166.2	124.57	920.1	98.28	30.0
50.0	282.8	527.2	1168.4	138.27	915.4	108.3	35.0
55.0	289.0	480.6	1170.1	151.91	910.9	118.3	40.0
60.0	294.7	441.6	1171.9	165.56	906.9	128.1	45.0
65.0	300.0	408.7	1173.5	179.13	903.0	137.8	50.0
70.0	305.0	380.4	1175.0	192.71	899.4	147.5	55.0
75.0	309.8	355.8	1176.5	206.29	896.0	157.1	60.0
80.0	314.3	334.3	1177.8	219.84	892.7	166.6	65.0
85.0	318.4	315.2	1179.1	233.38	889.8	176.1	70.0
90.0	322.4	298.2	1180.3	246.94	886.9	185.4	75.0
95.0	326.2	283.0	1181.5	260.46	884.2	194.9	80.0
100.0	329.9	269.4	1182.7	273.93	881.6	204.2	85.0
105.0	333.3	257.0	1183.7	287.40	879.1	213.4	90.0
110.0	336.8	245.7	1184.7	300.87	876.5	222.6	95.0
115.0	340.0	235.3	1185.7	314.33	874.2	231.8	100.0

HEAT: ITS MECHANICAL EQUIVALENT.

Heat is dynamic work, or the product of the three simple elements, —*force*, *velocity*,

and *time;* in which the temperature of the heat represents *force*, and the cubic contents of the units of heat represent the product of *time* and velocity, which is *space.*

It has been shown that *the change of heat equal to one pound of water raised in temperature one degree is one unit of heat.*

For water, the quantity of heat, measured by *heat units*, corresponding to any change of heat, may then be represented by the number of pounds of water multiplied by the number of degrees of the thermometer, which indicates the change of temperature.

Joule, in his experiments, found that one unit of heat developed 772 foot-pounds of work; water being taken at a temperature of 39.1° Fah., its point of maximum density.

The English unit of *dynamic work* is one pound lifted twelve inches, or one pound of force acting through one foot of space, and is called the foot-pound; 550 per second, or 33,000 foot-pounds or units of work, performed in one minute, make a horse-power.

The French unit of work is one kilo-

gram lifted one metre, called the kilogram-metre, or, for brevity, kilo-metre. It is equal to 7.233136 foot-pounds. Seventy-five kilogram-metres, exerted in one second, constitute the French horse-power, equal to 32,549,112 foot-pounds per minute.

The English horse-power is therefore 1.01416 French horse-powers, and the French horse-power is 0.986 of an English horse-power.

One horse-power will consume, or generate, 2,564 calorics (or heat) per hour.

Table No. 2.

COMPARISON OF DIFFERENT UNITS OF HEAT AND WORK.

English Calorics.		French Caloric.		Prussian	Dynamic Work.	
Fah. pounds.	Cent. pounds.	Fah. kilo.	Cent. kilo.	Cent. p. ft.	Foot pounds.	Kilo-metres.
1	0.5555	0.4536	0.252	0.5769	772.0	106.51
1.8	1	0.8165	0.4536	1.0385	1389.6	191.71
2.2047	1.2248	1	0.555	1.2719	1702.0	384.066
3.968	2.2047	1.8	1	2.2894	3063.6	626.52
1.733	0.9630	0.7862	0.4368	1	1368.2	273.66
0.0012953	0.0007196	0.0005876	0.000326	0.0007473	1	0.13823
0.0093896	0.0005205	0.0004250	0.0002361	0.0005405	7.233	1

MECHANICAL POWER.

Mechanical power, or work, is pressure acting through space; and the law of the

conservation of force teaches that power, once produced, cannot be annihilated, though it may be transformed into other forces of equivalent value. In all machines a certain proportion of the power resident in the prime mover is lost; while the rest is utilized, and is rendered available for the performance of those labors for which power is required. Thus, in a water-wheel, the theoretical value of the fall is that due to a certain weight of water falling through a certain number of feet in the minute; and if we know the height of the fall, and the discharge of water in a given time, the theoretical value of such a fall can be easily computed. But by no machine, whether a water-wheel or a turbine, can the whole of the power be extracted from the fall, and be made available for useful purposes. About *eighty per cent* of the theoretical power of a water-fall is considered to be a very satisfactory result to obtain in practice; and the rest is lost by impact and eddies, and by the friction of the water and of the machine.

In the steam-engine, the motive force is

not gravity, but heat; and just in the same way as power is imparted by water in descending from a higher to a lower level, so is power imparted by heat in descending from a higher to a lower temperature. These two temperatures are the temperature of the boiler, and the temperature of the atmosphere or condenser; and it is clear, that, if the atmosphere or condenser were as hot as the boiler, there would be no motion to the engine. And just as in a water-fall there is a certain theoretical power due to the quantity of gravitating matter and the difference of level, so in a steam-engine there is also a certain theoretical power due to the quantity of heated matter and the difference of temperature. But, in utilizing the power of steam-boilers, this theoretical limit is not approached so nearly as in hydraulic machines. In the steam-boiler, the larger part of the attainable fall of temperature is lost. Thus, if we suppose the temperature of the furnace to be 2,000° Fah., and the temperature of the boiler to be 300° (corresponding to 50 pounds per square inch above the atmos-

phere), while that of the condenser is 100°, we utilize pretty effectually the power represented by the difference in temperature between 100° and 300°; but the difference between 300° and 2,000° is not utilized at all. The consequence of this state of things is that not above *one-tenth* of the power, theoretically due to the fuel consumed, is utilized in the best modern steam-engines, — the rest being thrown away.

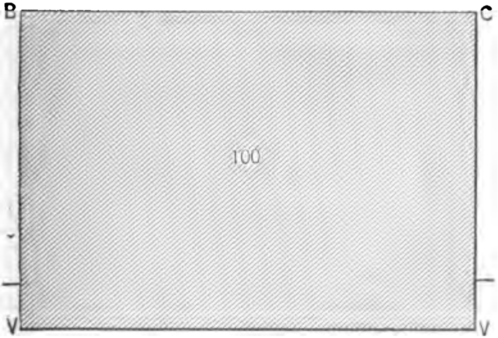
THE WORK OF EXPANSION.

When the valve for admission of steam to an engine-cylinder is open during the full stroke of the piston, the cylinder is filled with steam at every stroke, of a pressure nearly equal to that of the boiler. See Fig. 1.

In order to save steam, — or, more correctly, to employ its effect to a higher degree, — the admittance of steam to the cylinder is cut off when the piston has moved a portion of its stroke. From the cut-off point the steam acts expansively, with a decreased pressure on the piston, as shown in diagram (Fig. 2).

If we admit steam of $85 + 15 = 100$ pounds total pressure per square inch into a cylinder, and the valve closes the steam-port when the piston has travelled half the length of the stroke, from *B* to *e*, the steam remaining in the cylinder will expand to double its volume in forcing the piston to

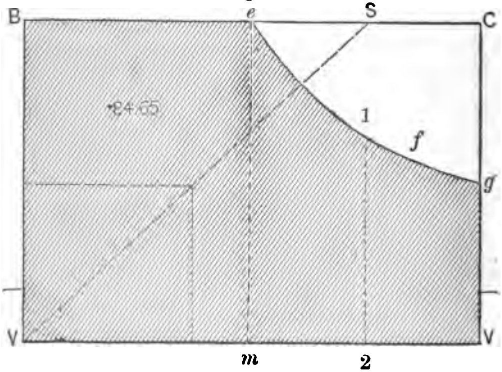
Fig. I



the end of the cylinder; and a certain amount of *work* has been done with half the quantity of steam, as in case of Diagram 2; and the steam, in expanding after the port was closed during the rest of the stroke, continues to do work, as the pressure of the expanding steam is greater than

that of the condenser. Now, this work performed after the steam was cut off is greatly in excess of that performed in Fig. 2, as compared to their respective volumes (as 5 is to 10), and has been obtained by the use of expansion. In this latter case

Fig.2



the steam expanded twice its volume, and its pressure was exactly half what it was before ; namely, 50 pounds per square inch.

In making this calculation for pressure of steam after it has expanded, the *total pressure*, *P*, must be used, which is reckoned from perfect *vacuum*.

In Fig. 1, *VB* is the diameter, and *VV* the length, of the cylinder; the pressure during the stroke, when there is no expansion, is assumed at 85 pounds, as per steam-gage, *plus* 15 pounds for perfect vacuum, equals 100 pounds total pressure per square inch.

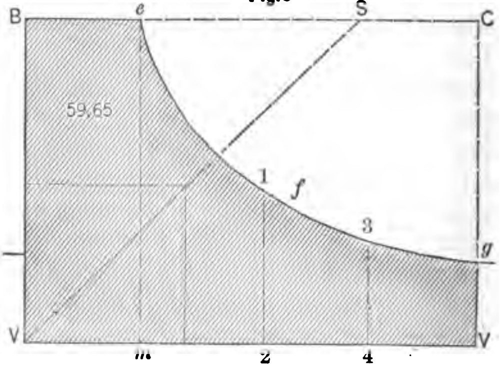
Now, if the steam is cut off when the piston has moved one-half the length of the cylinder (see Fig. 2), from *B* to *e*, the steam, whose volume is *V*, *B*, *e*, and *m*, must expand and fill the whole cylinder, its pressure getting less and less: so that such lines as *em*, 1 2, 3 4, and *gV*, in Figs. 2 and 3, represent pressures at different parts of the stroke, and the curve *e*, 1, 3, *f*, and *g* is the expansion curve.

Diagram Fig. 3 represents the same engine cutting off at one-quarter the stroke, the average pressure being 59.65 pounds mean pressure.

In fact, Figs. 1, 2, and 3 are imaginary indicator diagrams, supposed to be taken from a condensing engine (the average pressure from a non-condensing engine would be arrived at in the same way, but

15 pounds would be deducted after the calculations were made, to allow for pressure of the atmosphere); and hence their areas indicate the relative amounts of work per-

Fig.3



formed in a single stroke of the engine, when there is: —

First, No cut off.

Second, Cut off at half stroke.

Third, Cut off at one-fourth of the stroke.

Now, the area of Fig. 2 is nearly equal to that of Fig. 1; so that, when expansion is allowed, a cylinder *half full of steam* will

perform more than *three-fourths* as much work as the cylinder *full of steam*, at the *same initial pressure*, can perform *without expansion.*

As a further illustration, Fig. 3 is the diagram that would be made if the steam were cut off after the piston had travelled one-fourth of the stroke. In this case, only one-fourth the steam would be required, as was for Fig. 1, performing more than one-half as much as the latter with one-fourth of the steam.

Assuming that, in the cylinder, the volume of *steam varies inversely as the pressure*, the work done in one stroke of the piston is: —

$$AlP(1+x) \quad . \quad . \quad . \quad . \quad . \quad (1).$$

Or, in words, the work done is the value of the hyperbolic logarithm x, from Table 4, p. 47, plus one, multiplied by the product of the initial pressure P, multiplied by l (the distance the piston moves before steam is cut off), and this product by the area, A, of the cylinder in square inches.

Where A = area of cylinder, or piston, in square inches.

L = length of stroke of piston in inches.

l = distance travelled by the piston before the steam is cut off.

g = grade, or ratio, of expansion, $\frac{L}{l}$.

x = hyperbolic logarithm of g (see Table No. 4).

P = initial pressure of steam in pounds per square inch, measuring from perfect vacuum in cylinder before cut-off takes place.

p = Mean average pressure after cut off takes place and during full stroke, in pounds per square inch; and is found by the following formula:—

$$p = \frac{P}{g}(1 + x) \quad . \quad . \quad . \quad . \quad . \quad . \quad (2).$$

HYPERBOLIC LOGARITHMS.

In estimating the power which an engine will exert with a given pressure of steam, to be cut off at any given point of the stroke, we ascertain the mean pressure on the square inch which will be exerted during the stroke, by means of the table of hyperbolic logarithms, which latter are calculated for expansion according to the law of Boyle and Mariotte.

Table No. 3.

HYPERBOLIC LOGARITHMS.

No.	Logarithms.	No.	Logarithms.	No.	Logarithms.	No.	Logarithms.
0.0	0.00000	4.0	1.38629	7.0	1.94591	10	2.30258
1.1	0.09530	4.1	1.41096	7.1	1.96006	11	2.39589
1.2	0.18213	4.2	1.43505	7.2	1.97406	12	2.48491
1.3	0.26234	4.3	1.45859	7.3	1.98787	13	2.56494
1.4	0.33646	4.4	1.48161	7.4	2.00149	14	2.63906
1.5	0.40505	4.5	1.50408	7.5	2.01490	15	2.70805
1.6	0.46998	4.6	1.52603	7.6	2.02816	16	2.77259
1.7	0.53063	4.7	1.54753	7.7	2.04115	17	2.83321
1.8	0.58776	4.8	1.56859	7.8	2.05415	18	2.89037
1.9	0.64181	4.9	1.58922	7.9	2.06690	19	2.94444
2.0	0.69315	5.0	1.60944	8.0	2.07944	20	2.99573
2.1	0.74190	5.1	1.62922	8.1	2.09190	21	3.04452
2.2	0.78843	5.2	1.64865	8.2	2.10418	22	3.09104
2.3	0.83287	5.3	1.66770	8.3	2.11632	23	3.13549
2.4	0.87544	5.4	1.68633	8.4	2.12830	24	3.17805
2.5	0.91629	5.5	1.70475	8.5	2.14007	25	3.21888
2.6	0.95548	5.6	1.72276	8.6	2.15082	26	3.25810
2.7	0.99323	5.7	1.74046	8.7	2.16338	27	3.29584
2.8	1.02962	5.8	1.75785	8.8	2.17482	28	3.33220
2.9	1.06473	5.9	1.77495	8.9	2.18615	29	3.36730
3.0	1.09861	6.0	1.79175	9.0	2.19722	30	3.40120
3.1	1.13140	6.1	1.80827	9.1	2.20837	31	3.43399
3.2	1.16314	6.2	1.82545	9.2	2.21932	32	3.46574
3.3	1.19594	6.3	1.84055	9.3	2.23014	33	3.49651
3.4	1.22373	6.4	1.85629	9.4	2.24085	34	3.52636
3.5	1.25276	6.5	1.87180	9.5	2.25129	35	3.55535
3.6	1.28090	6.6	1.88658	9.6	2.26191	36	3.58352
3.7	1.30834	6.7	1.90218	9.7	2.27228	37	3.61092
3.8	1.33046	6.8	1.91689	9.8	2.28255	38	3.63759
3.9	1.36099	6.9	1.93149	9.9	2.29171	39	3.66356

MEAN PRESSURE.

When the steam is expanded in the cylinder, the mean pressure p, throughout the stroke of piston, will be less than the initial pressure P. The mean pressure p,

during expansion, will be according to formula (2); or, in words: —

Rule. — Divide the initial pressure P, by the proportion, or grade, g, of the stroke during which the steam is admitted, and multiply the quotient by hyperbolic logarithm x, plus one (take the value of x from Table No. 4).

RATIO, OR GRADE, OF EXPANSION.

The proportion, or grade, g, of the stroke during which the steam is admitted, is found by dividing the length L in inches of the cylinder swept through by the piston, by the length l in inches of the space into which the steam is admitted.

Example. — Suppose the length of the stroke of a given engine be $L = 80$ inches, the initial pressure P, 90 pounds per square inch, and the steam to be cut off at $l = 20$ inches of the stroke: what will be the mean pressure?

$$\text{Formula (2)} = p = \frac{P}{g}(1 + x).$$

Grade, or ratio, g, $= \frac{80}{20} = 4$ grade, or ratio, of expansion.

Hyperbolic logarithm of $4 = 1.386$ (see x in Table No. 4).

Then we have

$$p = \tfrac{40}{4} \times (1+1.386) = 53.68 \text{ pounds},$$

the mean pressure required.

The initial pressure P, given above, is the total pressure, measured from perfect vacuum. To find the initial P, add the atmospheric pressure — in common practice an atmosphere is generally taken as 15 pounds (14.7 exact) per square inch — to the pressure p, shown by the steam-gage; and from the mean pressure found as above subtract the counter or back pressure, to effective mean pressure exerted. Thus, in the above case, the steam-gage is supposed to show a pressure p of $90 - 15 = 75$ pounds only; and, if the calculation is being made for a condensing, or "low-pressure," engine, the estimated loss from imperfect vacuum must be subtracted (not less than *four* pounds), and if for a non-condensing, or "high-pressure," engine, the pressure of the atmosphere, and also any estimated counter or back pressure above that, must be subtracted from 53.68 pounds, the mean pressure obtained by the calculation.

HYPERBOLIC LOGARITHMS.

The hyperbolic logarithm of a number is found by multiplying the common logarithm of the number by 2.30258509.

Example. — The common logarithm of 3 is 0.4771213, which, multiplied by 2.30258509, gives 1.09861, the hyperbolic logarithm.

And, also, the hyperbolic logarithm, multiplied by 0.43329448, gives the common logarithm.

The following, Table No. 3, contains the hyperbolic logarithms for numbers running from 1.11, the grade, or ratio, of $\frac{9}{10}$, or 0.9 cut off, up to $\frac{1}{10}$, or 0.1, representing $\frac{1}{10}$ cut off, which is considered sufficient for application to expansion of steam for all practical purposes.

EXPANSION OF STEAM, AND ITS EFFECT WITH EQUAL VOLUMES OF STEAM.

The theoretical economy of using steam expansively is as follows, the same volume of steam being expended in each case, and expanded to fill the increased spaces: —

Table No. 4.

Portion of stroke at which steam is cut off.	Grade, or ratio, of expansion.	Hyperbolic logarithm.	Mean pressure of steam during the whole stroke.	Percentage of gain in fuel, or power.
l	*g*	*x*	*p*	%
$\frac{1}{10}$, or 0.1	10.0	2.302	3.302	230.0
$\frac{1}{8}$, or 0.125	8.0	2.079	3.079	208.0
$\frac{1}{6}$, or 0.166	6.0	1.791	2.791	179.0
$\frac{2}{10}$, or 0.2	5.0	1.609	2.609	161.0
$\frac{1}{4}$, or 0.25	4.0	1.386	2.386	139.0
$\frac{3}{10}$, or 0.3	3.33	1.203	2.203	120.0
$\frac{1}{3}$, or 0.333	3.0	1.099	2.099	110.0
$\frac{3}{8}$, or 0.375	2.66	0.978	1.978	97.8
$\frac{4}{10}$, or 0.4	2.5	0.916	1.916	91.6
$\frac{1}{2}$, or 0.5	2.0	0.693	1.693	69.3
$\frac{6}{10}$, or 0.6	1.666	0.507	1.507	50.7
$\frac{5}{8}$, or 0.625	1.6	0.47	1.47	47.0
$\frac{2}{3}$, or 0.666	1.5	0.405	1.405	40.5
$\frac{7}{10}$, or 0.7	1.42	0.351	1.351	35.1
$\frac{3}{4}$, or 0.75	1.33	0.285	1.285	22.3
$\frac{8}{10}$, or 0.8	1.25	0.223	1.223	20.5
$\frac{7}{8}$, or 0.875	1.143	0.131	1.131	13.1
$\frac{9}{10}$, or 0.9	1.11	0.104	1.104	10.4

In Table No. 4 no deductions are made for a reduction of the temperature of the steam while expanding, or for loss by back pressure.

The same relative advantages follow in expansion, as above given, whatever may be the initial pressure of the steam.

The pressure of the atmosphere is to be included in calculating the expansion. It must, therefore, be deducted from the results in non-condensing engines. In condensing engines a deduction must be made for imperfect vacuum. This will amount to about ($2\frac{1}{2}$ pounds per square inch) 5 inches in well-proportioned engines.

Where there is no cut-off, as in diagram Fig. 1, the work done equals APL, or the area, A, of the cylinder, multiplied by the absolute pressure P, and this product by the length of stroke, L, of the piston in feet.

When the cut-off takes place at one-fourth of the stroke L, at point e, diagram Fig. 3, there is only one-fourth as much steam admitted as in case of diagram Fig. 1; but the work, instead of being $\frac{APL}{4}$, or

$\frac{1 \times 100 \times 1}{4} = 25$, will be, as before stated, $AlP(1 + x)$, or $1 \times 0.25 \times 1(1 + 1.386) = 59.65$.

To make this more clear to the student and general reader, we will assume an engine doing actual work. It is well known that the most convenient way of calculating the horse-power of an engine is to multiply the area of the cylinder in square inches by the speed of the piston in feet per minute, and divide the product by 33,000. The result so obtained will be the horse-power of *one pound mean effective pressure*, and is called the horse-power constant, which, if multiplied by the whole mean effective pressure on the piston during the stroke, will give the indicated horse-power of the engine.

For example, suppose that the engine that would produce indicator diagrams as represented by Figs. 1, 2, and 3, had a stroke $L = 3$ feet, making 100 revolutions per minute, and a diameter of cylinder $A = 110$ square inches, and a piston speed of $100 \times 3 \times 2 = 600$ feet per minute: then

the horse-power value of one pound mean effective pressure will be as follows : —

$$\text{Horse-power constant} = \frac{110 \times 600}{33000}$$
$$= 2 \text{ horse-power.}$$

Now, diagram Fig. 1 averaged initial pressure $P = 100$ pounds, or a mean effective pressure throughout the stroke. The horse-power, therefore, will be as follows : —

$$\text{Horse-power} = 100 \times 2 = 200 \text{ horse-power.}$$

In diagram Fig. 3, the steam was cut off after the piston had moved from B to e, or one-fourth of its stroke; the grade, or ratio, of expansion being $\frac{36}{9} = 4$. Therefore, the mean effective pressure p will be, according to formula (2), —

$$p = \frac{P}{g}(1 + x);$$

or, substituting values,

$$p = \frac{100}{4}(1 + 1.386) = 59.65 \text{ pounds};$$

or, to simplify it still further, it will be as follows : —

$\frac{36}{9} = 4$ hyper. logarithm of $4 = 1.386 + 1 = 2.386$.

Then

$$\tfrac{100}{4} = 25 \times 2.386 = 59.65 \text{ pounds.}$$

Now, diagram Fig. 3 shows a mean effective pressure of 59.65 pounds, which, multiplied by the horse-power constant, will be, —

$$P = 59.65 \times 2 = 119.30 \text{ horse-power.}$$

Therefore, we see that one-fourth of the steam expanded performs *three-fifths*, or nearly *sixty per cent*, of the whole work; so that, by using expansion, the work obtained from one pound of steam is 2.386 times what was obtained when following full stroke, as shown by diagram Fig. 1, or a gain of *forty per cent* by using steam expanding *three-fourths* of the stroke.

$$\% = \frac{200 - 119.30}{200} = 40.5 \text{ per cent gain.}$$

The number 2.386 has lately been called the "indicator co-efficient" of the engine. By cutting off at *one-tenth* of the stroke, the efficiency of the steam is increased 3.3 times; that is, the "indicator co-efficient" is 3.3.

Expansion is valuable in another way.

At the end of every stroke the piston stops momentarily, returning on its old path; and it is advisable to prepare for the sudden reversal of motion of the piston, by diminishing the steam pressure. Now, when expansion is used, the greatest pressure is exerted at the beginning of the stroke, when the piston moves slowly, and when it is most advisable to get up a great velocity. The pressure after cut off diminishes gradually until it is very little greater than that of the atmosphere; so that the steam experiences little difficulty in escaping by the exhaust-passages on the return stroke. In fast-running engines, the exhaust-port is opened before the end of the stroke, and the exhaust-port on the other side of the piston is closed, that there may be a cushion of the steam to prevent "shocks" or "jars."

THE PERFORMANCE OF STEAM WHEN EXPANDED.

Steam, in its ordinary condition as saturated steam, though it does not rank as a perfect gas, nevertheless acts in the cylin-

der of a steam-engine so much to the same effect as a perfect gas could do, that its performance may be treated in the same way as if it were perfect as a gas. The quality in consideration of which a gas is said to be perfect is, as has already been stated, its property of expanding into a larger volume in the same proportion inversely as the pressure falls, the temperature being supposed to remain the same. Now, though saturated steam does not and can not exactly follow this ratio, seeing that the pressure falls more rapidly than the volume increases, yet it is found that the work performed by steam by expansion in the cylinder of an engine is practically the same as if it acted on the principle of a perfect gas.

Therefore it will be seen that the curve described by the pencil of an indicator, indicating the falling pressure of dry saturated steam expanding behind an advancing piston, is, if not exactly, nearly hyperbolic in its nature, or such that the products of the pressures at all points of the stroke, multiplied by the respective volumes of steam, are equal to each other.

Table No. 5.

MEAN AND INITIAL PRESSURE IN THE CYLINDER.

[Assuming that the pressures are inversely as the volume.]

Portion of stroke at which steam is cut off.	Grade, or ratio, of expansion.	Hyperbolic logarithm.	Mean pressure during stroke, the initial pressure being taken at 1.	Initial pressure in cylinder, the mean pressure being taken as 1.
l	g	x	p	P
$\frac{3}{4}$, or 0.75	1.333	0.2876	0.965	1.036
$\frac{7}{10}$, or 0.7	1.428	0.3506	0.949	1.054
$\frac{2}{3}$, or 0.666	1.5	0.4055	0.937	1.067
$\frac{6}{10}$, or 0.6	1.666	0.5108	0.904	1.106
$\frac{1}{2}$, or 0.5	2.0	0.6931	0.846	1.182
$\frac{4}{10}$, or 0.4	2.5	0.9163	0.766	1.305
$\frac{1}{3}$, or 0.333	3.0	1.0986	0.669	1.495
$\frac{3}{10}$, or 0.3	3.333	1.2040	0.661	1.513
$\frac{1}{4}$, or 0.25	4.0	1.3863	0.596	1.678
$\frac{2}{10}$, or 0.2	5.0	1.6094	0.522	1.916
$\frac{1}{6}$, or 0.166	6.0	1.7918	0.465	2.150
$\frac{1}{7}$, or 0.142	7.0	1.9459	0.421	2.375
$\frac{1}{8}$, or 0.125	8.0	2.0795	0.385	2.598

Table No. 5—(continued).

Portion of stroke at which steam is cut off.	Grade, or ratio, of expansion.	Hyperbolic logarithm.	Mean pressure during stroke, the initial pressure being taken at 1.	Initial pressure in cylinder, the mean pressure being taken as 1.
l	*g*	*x*	*p*	*P*
$\frac{1}{9}$, or 0.111	9.0	2.1972	0.355	2.817
$\frac{1}{10}$, or 0.1	10.0	2.3025	0.330	3.030
$\frac{1}{11}$, or 0.09	11.0	2.3979	0.309	3.236
$\frac{1}{12}$, or 0.083	12.0	2.4849	0.293	3.448
$\frac{1}{13}$, or 0.076	13.0	2.5649	0.274	3.649
$\frac{1}{14}$, or 0.071	14.0	2.6391	0.260	3.846
$\frac{1}{15}$, or 0.066	15.0	2.7081	0.247	4.048
$\frac{1}{16}$, or 0.062	16.0	2.7726	0.236	4.237
$\frac{1}{17}$, or 0.058	17.0	2.8332	0.226	4.425
$\frac{1}{18}$, or 0.055	18.0	2.8904	0.216	4.629
$\frac{1}{19}$, or 0.052	19.0	2.9444	0.208	4.807
$\frac{1}{20}$, or 0.05	20.0	2.9967	0.200	5.0
$\frac{1}{21}$, or 0.047	21.0	3.0445	0.192	5.208
$\frac{1}{22}$, or 0.045	22.0	3.0910	0.186	5.376
$\frac{1}{23}$, or 0.043	23.0	3.1355	0.180	5.555
$\frac{1}{24}$, or 0.041	24.0	3.1781	0.174	5.747
$\frac{1}{25}$, or 0.04	25.0	3.2189	0.169	5.917

DIAGRAM OF WORK DONE DURING EXPANSION.

The hyperbolic curve of expansion, expressive of the falling pressure, relative to the increasing volume, is represented by *Dg* on diagram Fig. 4.

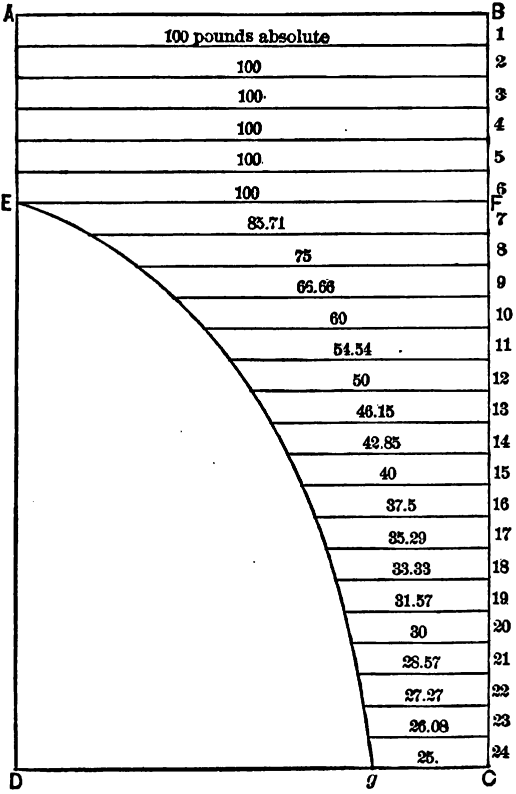
The rectangle *ABCD* is supposed to be the section of a cylinder, having a stroke of 24 inches. The diagram is divided into 24 parts, or inches of stroke. During six of these, that is, six inches of the stroke, or one-fourth, *AE*, the steam is admitted; and it is expanded during the remaining three-fourths, *FC*. Assuming that there is no clearance, the terminal pressure *gC* would be one-fourth of the initial pressure *P*, during admission; that is, it would be equal to the initial pressure *P*, taken in this case at 100 pounds total pressure per square inch, multiplied by the period of admission, and divided by the length, $l = 6$ inches of the stroke, or

$$100 \times \tfrac{6}{24} = 25 \text{ pounds per square inch,}$$

the terminal pressure.

The pressure for any intermediate point of the stroke may be found, similarly, by

Fig. 4.


A
B
100 pounds absolute
100
100
100
100
100
E
F
85.71
75
66.66
60
54.54
50
46.15
42.85
40
37.5
35.29
33.33
31.57
30
28.57
27.27
26.08
25.
1
2
3
4
5
6
7
8
9
10
11
12
13
14
15
16
17
18
19
20
21
22
23
24
D
g
C

taking the portion of the stroke described, from the commencement to the given point, as the divisor. Thus, at the end of 12 inches of the stroke, the total pressure is —

$$100 \times \tfrac{6}{12} = 50 \text{ pounds per square inch.}$$

Finding the pressure similarly for each intermediate inch of the stroke, and drawing ordinates for each inch of stroke, the curve may be formed by tracing it through the extremities of the ordinates, as shown in the figure.

The process of finding the intermediate pressures is, in fact, a case of proportion; and the following statements, showing the proportional process for each of the ordinates, make it quite clear: —

As 7 spaces : 6 spaces : : 100 lbs. pr. : 85.71 lbs. pr. at 7 inch.
8 spaces : 6 spaces : : 100 lbs. pr. : 75.00 lbs. pr. at 8 inch.
9 spaces : 6 spaces : : 100 lbs. pr. : 66.66 lbs. pr. at 9 inch.
10 spaces : 6 spaces : : 100 lbs. pr. : 60.00 lbs. pr. at 10 inch.
11 spaces : 6 spaces : : 100 lbs. pr. : 54.54 lbs. pr. at 11 inch.
12 spaces : 6 spaces : : 100 lbs. pr. : 50.00 lbs. pr. at 12 inch.
13 spaces : 6 spaces : : 100 lbs. pr. : 46.15 lbs. pr. at 13 inch.
14 spaces : 6 spaces : : 100 lbs. pr. : 42.85 lbs. pr. at 14 inch.
15 spaces : 6 spaces : : 100 lbs. pr. : 40.00 lbs. pr. at 15 inch.
16 spaces : 6 spaces : : 100 lbs. pr. : 37.50 lbs. pr. at 16 inch.
17 spaces : 6 spaces : : 100 lbs. pr. : 35.29 lbs. pr. at 17 inch.
18 spaces : 6 spaces : : 100 lbs. pr. : 33.33 lbs. pr. at 18 inch.
19 spaces : 6 spaces : : 100 lbs. pr. : 31.57 lbs. pr. at 19 inch.

20 spaces : 6 spaces : : 100 lbs. pr. : 30.00 lbs. pr. at 20 inch.
21 spaces : 6 spaces : : 100 lbs. pr. : 28.57 lbs. pr. at 21 inch.
22 spaces : 6 spaces : : 100 lbs. pr. : 27.27 lbs. pr. at 22 inch.
23 spaces : 6 spaces : : 100 lbs. pr. : 26.08 lbs. pr. at 23 inch.
24 spaces : 6 spaces : : 100 lbs. pr. : 25.00 lbs. pr. at 24 inch.

Having got so far, the work done by expansion may be calculated from these particulars without the aid of hyperbolic logarithms.

THE THEORETICAL POSSIBILITY OF GAIN BY EXPANSION.

To find the increase of efficiency arising from using steam expansively: —

Rule. — Divide the total length of the stroke by the distance (which call 1) through which the piston moves before the steam is cut off. The Napierian logarithm of the part of the stroke performed with the full pressure of steam before cut off represents the increase of efficiency due to expansion.

Example. — Suppose that the steam be cut off at $\frac{2}{10}$ (two-tenths), or 0.2, of the stroke: what is the increase of efficiency due to expansion?

Now, 0.2 of the whole stroke is the same

as $\frac{1}{5}$ of the whole stroke; and the ratio, or grade, of expansion equals 5. The hyperbolic logarithm of 5 is 1.609, which, increased by 1, the value of the portion performed with full initial pressure, gives $1.609 + 1 = 2.609$ as the relative efficacy of the steam when expanded to this extent (eight-tenths), instead of 1, which would have been the efficacy if there had been no expansion.

If the steam be cut off at $\frac{1}{10}$, $\frac{2}{10}$, $\frac{3}{10}$, $\frac{4}{10}$, $\frac{5}{10}$, $\frac{6}{10}$, $\frac{7}{10}$, $\frac{8}{10}$, or $\frac{9}{10}$ of the stroke, the respective ratios, or grades, of expansion will be 10, 5, 3.33, 2.5, 1.66, 1.42, 1.25, and 1.11; of which numbers the respective hyperbolic logarithms are 2.303, 1.609, 1.203, 0.916, 0.693, 0.507, 0.351, 0.223, 0.104: and, if the steam be cut off at $\frac{1}{8}$, $\frac{2}{8}$, $\frac{3}{8}$, $\frac{4}{8}$, $\frac{5}{8}$, $\frac{6}{8}$, or $\frac{7}{8}$ of the stroke, the respective ratios, or grades, of expansion will be 8, 4, 2.66, 2, 1.6, 1.33, and 1.14; of which the respective hyperbolic logarithms are 2.079, 1.386, 0.978, 0.693, 0.470, 0.285, 0.131. With these data, it will be easy to compute the mean pressure of steam of any given initial pressure when cut off at any eighth

part or any tenth part of the stroke; as we have only to divide the initial pressure of the steam in pounds per square inch by the ratio of expansion, and to multiply the quotient by the hyperbolic logarithm, increased by 1, of the number representing the ratio, or grade, which gives the mean pressure throughout the stroke in pounds per square inch. Thus, if steam of $65 + 15 = 80$ pounds absolute be cut off at half-stroke, the ratio, or grade, of expansion is 2; and 80 divided by $2 = 40$, which, multiplied by $1 + 0.693 = 67.72$, which is the mean pressure throughout the stroke in pounds per square inch. The terminal pressure is found by dividing the initial pressure by the ratio, or grade, of expansion; thus, the terminal pressure of steam of 80 pounds cut off at half-stroke will be 80 divided by $2 = 40$ pounds per square inch.

Example. — What will be the mean pressure, throughout the stroke, of steam of 160 pounds per square inch cut off at $\frac{1}{8}$ the stroke?

Divide 160 by $8 = 20$, which, multiplied by 3.079 (the hyperbolic logarithm of 8

increased by 1, $2.079 + 1 = 3.079$), gives 61.58, which is the mean pressure exerted on the piston throughout the stroke, in pounds per square inch. If the steam were cut off at $\frac{1}{10}$ of the stroke, instead of $\frac{1}{8}$, then we should have 160 divided by $10 = 16$, which, multiplied by 3.303 (the hyperbolic logarithm of 10 increased by $1 = 2.303 + 1 = 3.303$), gives 52.85 pounds, which would be the mean pressure on the piston throughout the stroke in such a case.

If the initial pressure of the steam were 10 pounds per square inch, and the expansion took place throughout $\frac{8}{10}$ of the stroke, or the steam were cut off at $\frac{2}{10}$, then 10 divided by $5 = 2$, which, multiplied by $2.600 = 5.218$ pounds per square inch of mean pressure.

GAIN IN FUEL BY EXPANSION.

When steam is cut off before the end of the stroke in a cylinder, the pressure effected by it for the portion at which it flowed for full stroke is represented by 1; and the pressure exerted afterward, by the result due to the relative expansion. The total pressure, or work, is represented by

the sum of these units. If the steam had flowed for the full stroke of the piston, the pressure would have been 1 added to the proportionate distance for which the steam was expended in case of expansion.

The gain of working steam expansively is as follows: —

Cutting off at $\frac{1}{10}$ the stroke, efficacy is increased 3.300 times.
Cutting off at $\frac{1}{8}$ the stroke, efficacy is increased 3.000 times.
Cutting off at $\frac{2}{10}$ the stroke, efficacy is increased 2.600 times.
Cutting off at $\frac{1}{4}$ the stroke, efficacy is increased 2.386 times.
Cutting off at $\frac{3}{10}$ the stroke, efficacy is increased 2.200 times.
Cutting off at $\frac{3}{8}$ the stroke, efficacy is increased 1.980 times.
Cutting off at $\frac{4}{10}$ the stroke, efficacy is increased 1.920 times.
Cutting off at $\frac{1}{2}$ the stroke, efficacy is increased 1.690 times.
Cutting off at $\frac{6}{10}$ the stroke, efficacy is increased 1.500 times.
Cutting off at $\frac{5}{8}$ the stroke, efficacy is increased 1.470 times.
Cutting off at $\frac{7}{10}$ the stroke, efficacy is increased 1.350 times.
Cutting off at $\frac{3}{4}$ the stroke, efficacy is increased 1.280 times.
Cutting off at $\frac{8}{10}$ the stroke, efficacy is increased 1.220 times.
Cutting off at $\frac{7}{8}$ the stroke, efficacy is increased 1.130 times.
Cutting off at $\frac{9}{10}$ the stroke, efficacy is increased 1.100 times.

Table No. 6 shows the value of the portion of steam after cut off; its relative efficacy by expansion being as this indicates, instead of being *one*, which would have been the efficacy had there been no expansion. From the above we can compute the gain in fuel as follows: —

Rule. — Divide the relative effect, or, in other words, the number of times the efficacy is increased, by the grade of expansion g (see table of hyperbolic logarithms), and divide 1 by the quotient. The result is the initial pressure of steam required to be expanded to produce a like effect of steam at full stroke. Divide this pressure by the number of times the steam is expanded, and subtract the quotient from 1. The remainder will give the percentage of gain of fuel.

Example. — Suppose the steam in an engine to be cut off after the piston has moved one-fourth the length of the stroke, what is the gain in fuel?

The relative effect (see efficacy due to expansion, on p. 63) equals 2.386, and the number of times of expansion equals 4. Then

$$2.386 \div 4 = 0.5965,$$

and

$$1.00 \div 0.5965 = \text{initial pressure} = 1.64,$$

and

$$1.64 \div 4 = 0.41,$$

and

$$1.00 - 0.41 = \text{per cent} = 59.$$

Table No. 6.

MEAN PRESSURE OF EXPANDING STEAM.

Absolute steam pressure per square inch.	GRADE OF EXPANSION OF STEAM, DENOTED BY x.						
	4	3	2.666	2	1.6	1.5	1.333
	Steam cut off at l, from beginning of stroke.						
P	$\frac{1}{4}$ or 0.25	$\frac{1}{3}$ or 0.333	$\frac{3}{8}$ or 0.375	$\frac{1}{2}$ or 0.50	$\frac{5}{8}$ or 0.625	$\frac{2}{3}$ or 0.666	$\frac{3}{4}$ or 0.75
20	11.931	13.991	14.853	16.931	18.350	18.734	19.304
25	14.913	17.488	18.567	21.164	22.938	23.481	24.130
30	17.897	20.986	22.280	25.396	27.524	28.100	28.956
35	20.880	24.484	25.992	29.630	32.110	32.784	33.782
40	23.860	27.964	29.670	33.860	36.750	37.333	38.550
45	26.842	31.459	33.378	38.092	41.341	42.000	43.368
50	29.828	34.977	37.133	42.328	45.875	46.835	48.262
55	32.811	38.474	40.846	46.561	50.462	51.518	53.088
60	35.794	41.972	44.559	50.794	55.050	56.202	57.914
65	38.777	45.470	48.273	55.027	59.637	60.885	62.740
70	41.760	48.967	51.986	59.260	64.225	65.569	67.566
75	44.743	52.465	55.700	63.493	68.812	70.252	72.393
80	47.726	55.963	59.413	67.726	73.400	74.936	77.216
85	50.709	59.461	63.126	71.959	77.987	79.619	82.042
90	53.692	62.958	66.840	76.192	82.574	85.303	86.866
95	56.675	66.456	70.553	80.425	87.163	89.986	91.699
100	59.657	69.954	74.267	84.657	91.750	93.670	96.524
105	62.640	73.451	77.981	88.890	96.337	98.353	101.35
110	65.622	76.949	81.694	93.123	100.92	103.04	106.17
115	68.606	80.447	85.407	97.356	105.51	107.72	111.00

TERMINAL PRESSURE.

Rule for finding the pressure at the end

of the stroke, or at any point during expansion: —

P = initial pressure of steam in pounds per square inch, including the pressure of the atmosphere.
L = distance travelled by the piston when the pressure of steam = x.
l = distance travelled by the piston before the steam is cut off.
x = pressure of steam in the cylinder, including the pressure of the atmosphere when the piston has travelled a distance L.

$$x = \frac{Pl}{L}.$$

Or, in words, the terminal pressure for any cut off is the absolute pressure P, multiplied by the distance l, the piston has moved when steam is cut off, and this product divided by stroke L.

The steam pressure on the boiler is readily known; but the steam in its passage to the cylinder is subject to various losses, as "wire-drawing," condensation, friction, etc., so that frequently the pressure on the piston does not exceed two-thirds of that on the boiler.

Therefore, recourse must be had to the indicator for furnishing the exact data for ascertaining the exact pressure in the cylinder, so as to ascertain the power exerted by the engine, namely, the *mean* or *average* pressure of steam ; or, more accurately, the excess of pressure on the acting side of the piston to produce motive force. And from no other source can it be accurately ascertained.

In every branch of science our knowledge increases as the power of measurement becomes improved; and we have now to discuss the measuring instrument peculiarly appropriated to the steam-engine, namely, the indicator invented by Watt. The student must thoroughly understand the reading of an indicator diagram before he can appreciate the reason for the various methods of construction adopted with reference to some of the working parts of an engine.

THE STEAM-ENGINE INDICATOR, AND ITS USE.

THE indicator is an instrument for showing the pressure of steam in the cylinder at all points in the stroke, and registering the varying pressures, as the piston moves to and fro, on a piece of paper secured to a revolving drum, by a pencil attached to the indicator piston.

The indicator was invented and first used by James Watt. For some time it was kept by him a secret, but became known before his death; and to its use, now quite general, we are more indebted than to any thing else for the degree of excellence which the steam-engine has attained.

The improved forms of instruments, known as the "Thompson," "Crosby," and "Tabor" indicators, are now largely employed. Each consists of a small cylinder accurately bored out, and fitted with a piston capable of working in the cylinder

with little or no friction, and yet practically steam-tight. The piston-rod is attached to a pair of light levers, so linked together that a pencil, carried at the end of one of the levers, moves in nearly a straight line vertically. A spiral spring placed in the cylinder above the piston, and of a strength proportioned to the steam pressure, resists the motion of the piston; and the elasticity of this spring is such that each pound pressure on the piston causes the pencil to move a certain fractional part of an inch.

The paper is wound round the drum, parallel to, and connected by a bracket with, the cylinder of the instrument, which has a diameter of two inches, and is capable of a semi-rotatory motion upon its axis of such an extent that the extreme length of diagram may be five inches. Motion is given to the drum in one direction by means of a cord connected with a suitable part of the engine, revolving the same way with that of the piston of the engine; and the drum makes its return movement by the action of a coiled spring at the base of the drum.

The indicator may be fixed in any de-

sired position; and the guide-pulleys, attached to the instrument at the base of the paper-drum, may also be placed so as to bring the cord upon the drum-pulley from any convenient direction. The upper end of the piston is open to the atmosphere; the lower end may, by means of a three-way cock on it, be put into communication either with the atmosphere or the cylinder of the engine. When the key of the cock is so turned that both sides of the indicator piston are acted on by the atmosphere, the pencil, on being brought into contact with the moving paper, will rule a straight line. This is called the atmospheric line.

It is now a common practice to connect, by means of pipes, both ends of the steam-engine cylinder (fitted with a three-way cock); so that, by one movement of the lever of the latter, the indicator can be put in communication with either end of the cylinder at pleasure, or it may be shut off from both, to produce the atmospheric line as above described.

In order to produce a diagram showing the action of the steam in the engine cylin-

der, if the indicator is put in connection with one end of the engine cylinder, the spring will be compressed by the steam pressure under it; and the amount to which the indicator piston rises is a measure of the steam pressure. For example, suppose that the spring compresses one-sixteenth of an inch for every pound on it: then, if the steam pressure is forty pounds, the piston will rise two and one-half inches. As the piston of the engine travels forward on its stroke, the steam pressure begins to diminish, and becomes less and less liable to compress the indicator spring; and consequently the indicator piston continually falls. In order to register these continually varying pressures, the pencil is kept in contact with the paper on the drum of the indicator; and, as the engine piston moves backwards and forwards, the drum of the indicator partially rotates also, backwards and forwards, coincident with that of the engine piston. The curved line thus traced by the pencil moving vertically up and down on the paper (itself moving at right angles to the up-and-down movement

of the pencil) is called an indicator card or diagram. It is nothing more than a register of the varying pressures in the cylinder as the piston moves to and fro.

In order, also, that the diagram shall be correct, it is essential, first, that the motion of the drum and paper shall coincide exactly with that of the engine piston; second, that the position of the pencil shall precisely indicate the pressure of steam in the cylinder.

The first condition is frequently somewhat difficult to bring about, because it is not only necessary that the beginning and end of the motions shall be coincident, but that these and all intermediate points shall be so. Owing to the irregular motion of the engine piston, consequent upon the varying angularity of the connecting-rod, it is, therefore, generally advisable to connect the cord in some way to the piston-crosshead. If any other point be chosen, it must be carefully seen that the motion given does not vitiate the diagram.

As the motion of the parts mentioned exceeds in length the motion of the indicator,

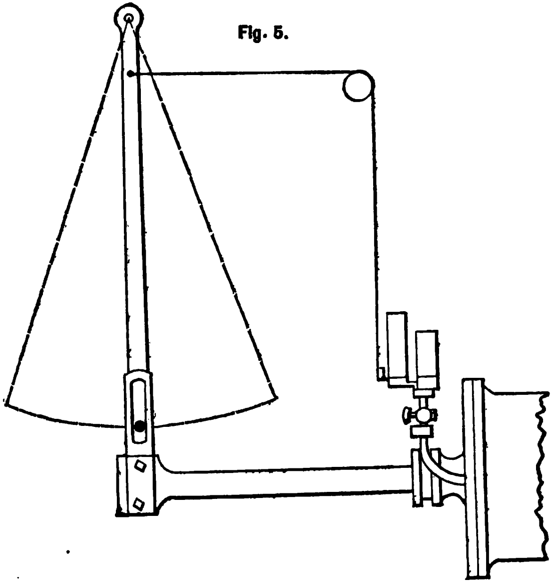
it must be reduced in length by levers of such proportions as may be required for that purpose. For example, if the stroke of the engine is thirty-six inches, and the length of the diagram is to be four inches, then the lengths of levers are as *one* is to *nine;* or, if only one lever is used, then the indicator motion must be taken from a point on the lever sufficiently far from its fixed end to obtain the reduced travel required.

Two of the simplest ways of reducing the motion are by a swinging lever, with a pin working in a slot of an arm secured to the crosshead of the engine, and transmitting the motion by a cord to the indicator, as shown on pp. 74, 75.

If the indicator and its spring are in good order, the pressure given by it may be taken as correct, provided the instrument is not placed too near, or exactly opposite, a steam-port. In the former case, the flow of steam past the opening into the indicator generally reduces the pressure indicated during the admission of the steam; and, in the latter case, the momentum of the steam,

especially if wet, causes the pencil to give jerky and uncertain indications. The cylinder-heads are usually the best position for the indicator.

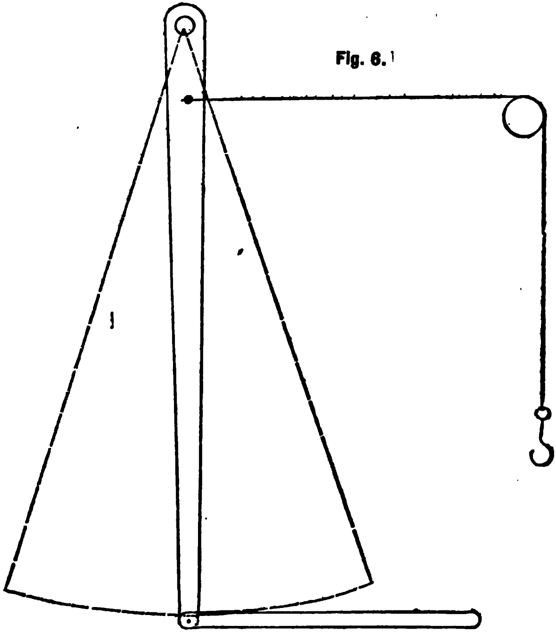
Fig. 5.



It is advisable to take diagrams from both ends of a cylinder, as the two diagrams always differ more or less; and, in

calculating the power, the area of the piston-rod must be deducted from the area of that side of the piston.

Fig. 6.



The indicator should be run for a few minutes, so that it may become hot, before

the diagrams are taken; and if any part works stiffly, this should be rectified. After taking the diagram, a note should be made, on the back, of the date, name of builder of engine, description of engine, diameter of cylinder, length of stroke, number of revolutions, the atmospheric pressure, boiler pressure, scale of diagram, and any other particulars it is desirable to have on record.

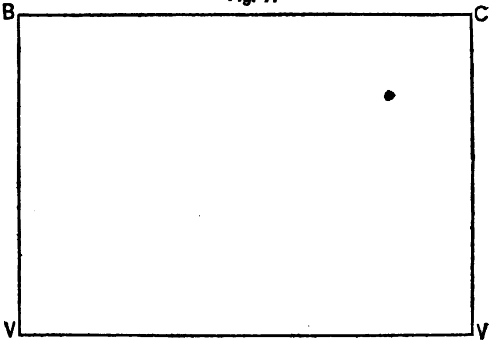
As I have before stated, the indicator is an instrument by means of which a steam-engine is caused to write on a piece of paper an accurate record of the performance that takes place within the cylinder. It gives a record which, to the uninstructed eye, is unintelligible, but by engineers is looked upon as the most reliable statement they can have of the work done by an engine, inasmuch as it tells at each and every part of the stroke of the piston what are the effective pressures tending to produce motion, and what are the back pressures tending to detract from the effective pressures.

INDICATOR DIAGRAMS.

Assuming that we have an indicator at-

tached to a steam-engine cylinder, and so connected that the drum containing the paper is moving to and fro coincident to the piston of the engine; before letting in steam to the indicator cylinder, if we apply the pencil to the surface of the paper, it

Fig. 7.



will draw upon the paper a horizontal line, *VV*, in length proportioned to the stroke of the engine. (See Fig. 7.)

Now, if we open the cock attached to the indicator cylinder, and assume that the engine piston has just commenced to move from left to right, the indicator piston will also move vertically; and the pencil will

trace the line *VB*, representing the pressure per square inch of the steam in the engine cylinder.

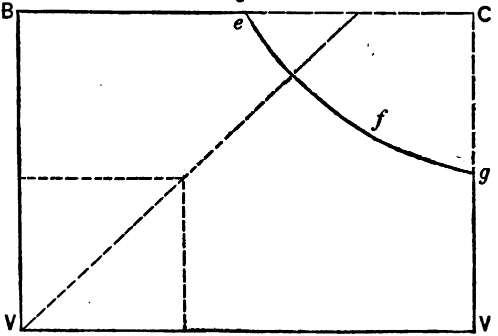
Assuming that the indicator spring be one which would compress one inch for every 40 pounds pressure per square inch acting on the piston, then, if there were 100 pounds pressure per square inch on the engine piston, the pencil would rise two and a half inches from *V* to *B*. Now, suppose the engine piston to have completed its stroke, —the pencil having traced the line *BC*, — and the slide-valve to have opened the exhaust-port so as to allow the steam to escape: then the indicator piston will fall, and the line *CV* will be traced. On the return stroke, the pencil would follow the line *VV*, with the exception of any diversion caused by steam that might remain in the cylinder in consequence of the steam not having been perfectly exhausted. Leaving this out of the question, it would have returned to the point *V* on the right, and thence to *V* on the left, thus describing a parallelogram, of which the horizontal line *VV* would represent a proportion of the

stroke of the piston, and the vertical line *VB* would represent the steam pressure upon the pistons. The area of this parallelogram would, therefore, represent pounds pressure into feet moved through by the piston in its stroke, or revolution of the engine.

Now, for simplicity, suppose that the line *VV* (Fig. 7) represents a foot-stroke of the piston of 1 foot, that the piston has an area of 99 square inches, and that the line *VB* represents 100 pounds pressure to the square inch: then we shall have 100 pounds multiplied by 1 foot; and this equals 100 foot-pounds, which, multiplied by 99 square inches (area), will equal 9,900 pounds as the work performed by the piston in one stroke, or half revolution. For both strokes, we have 9,900 multiplied by 2, equalling 19,800 pounds, as the force exerted by the engine through one revolution. If the engine makes 100 revolutions per minute, then 19,800 × 100 = 1,980,000 pounds would be the force exerted by the piston of such an engine in one minute. This, divided by 33,000, gives 60 horse-power, which is called the gross indicated horse-power.

Diagram, Fig. 7, is one that seldom, if ever, occurs in practice. When such are produced, they are only justified by the desire to obtain the greatest possible power from a given size of engine, without regard to the highest economy. It will be seen that steam was supposed to have been

Fig. 8.



admitted during the whole length of the stroke, and that no advantage whatever has been taken of the expansive property of the steam.

Diagram Fig. 8 shows steam used expansively.

Assume the same data as in former case,

— the 100 pounds pressure above the atmosphere has raised the pencil from *V* to *B;* also assume that the steam has been admitted to the engine cylinder up to the point *e* (half the length of the stroke), and then cut off by the valve: the steam now in the cylinder begins to expand; and, as it expands, it loses pressure. By the time, therefore, that the piston has arrived at *C* from *e*, the steam will have lost pressure; and the pencil will gradually fall, and trace the curved line *f*. By the time the piston has reached the end of the stroke, the pressure will further have diminished, say to *g;* and, when the exhaust opens, it falls down to *V*.

It will be seen by this diagram, that, although only half as much steam was admitted into the cylinder as in the case of diagram Fig. 7, the area of the diagram is very much more than half of that of Fig. 7. As a matter of fact, it is about 0.83 of that area; and thus a power 0.83 has been obtained by using expansively half the steam that was required in the case of Fig. 7.

As a further illustration, Fig. 9 is a dia-

gram that would be produced if the steam were cut off when the piston had moved one-fourth of the stroke. In this instance, only one-fourth the steam would be required as for Fig. 7. But the total area of the diagram is about 0.54 of that of

Fig. 9.

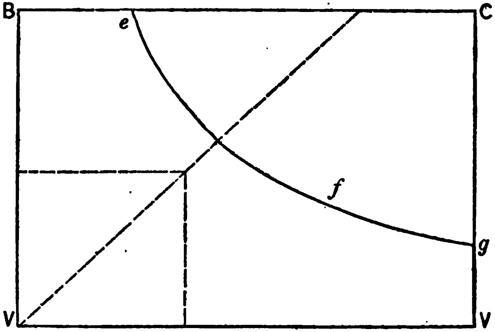


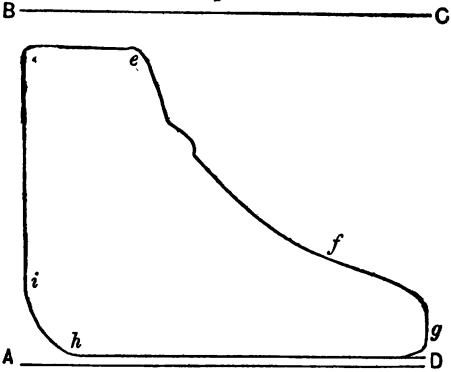
Fig. 7; so that 0.54, or more than one-half as much work, is obtained for one-fourth of the steam.

Fig. 10 is a diagram taken from a Corliss engine, 8 inches diameter and 24 inches stroke, 90 revolutions per minute.

Starting from the top corner *B*, the steam pressure remains uniform to about point *e*;

here, the cut-off valve being closed, the pressure commenced to fall, as represented by the curved line *f*, until it reached the point *g*, when the exhaust-valve being opened, allowing the steam to pass into the atmosphere, it quite suddenly drops from

Fig. 10.



g to *D*,—when the piston begins to return. There remains a slight pressure in the cylinder until the time the piston gets to *h*, that is the back-pressure throughout the stroke; so that it keeps the line of the pencil about 0.6 of a pound above the atmospheric line *AD*, until the closing of the

exhaust-valve, which occurs at the point *h*, — after which time the steam remaining in the cylinder is compressed, raising the indicator pencil, and forming the curved line *hi*.

In this case, the effective work done by the engine is represented by the area contained within the irregular figure, *B*, *e*, *f*, *g*, *D*, *h*, and *i*. This is after allowing for the back-pressure and the compression, which are contained between that figure and the lines *i*, *h*, and *D*.

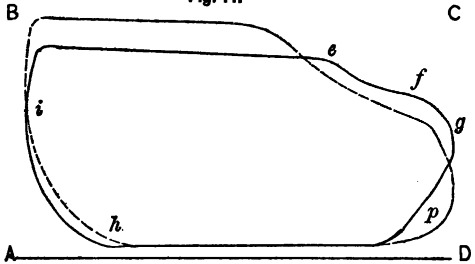
We have now described how a diagram is taken from one end of the cylinder. To obtain it from the other, all that has to be done is to make a pipe connection from the three cylinder-heads fitted with a two-way cock (as before described); and diagrams may be got on the same piece of paper, and would, if the engine were perfectly equal in performance at the two ends, be represented as it was in this case, by the dotted line on Fig. 9. The sum of these two areas will represent pounds pressure through the length of the stroke of the piston in a whole revolution, which, multiplied by the area of

the piston and the number of revolutions per minute, will give the foot-pounds. This, divided by 33,000, will give the gross indicated horse-power of the engine.

USE OF THE INDICATOR FOR SHOWING THE CONDITION OF THE ENGINE.

The indicator tells us not merely the power exerted by the engine, but the nature

Fig. 11.



of the faults by which the power is impaired. Thus, the shape of the indicator diagram may show that the steam or exhaust ports are too small, or that the valve has not sufficient lead, or is improperly set. Let us take, for example, the above diagram, Fig. 11.

When the indicator pencil is at the point

B, the engine piston is at the commencement of its stroke, the paper drum in motion. The line is traced from *B* to *e*, and thence to *g*, at which point the stroke is finished in this direction. At the point *e* the valve closed the steam-port, or, in other words, the steam was cut off; and, while the line from *e* to *g* was being traced, the steam pressure in the engine cylinder was expanding, and its pressure consequently decreasing, as shown by the falling of the line *f*. The line from *e* to *g* being convex, instead of concave (as per dotted line), shows that either the slide-valve or the piston, probably both, were not in good order, and admitted steam during expansion. The fall of the steam-line from *B* to *e* also shows that the steam-ports are too small. At the point *g* the exhaust-valve is opened to the atmosphere, the steam escapes, the pressure in the engine cylinder falls, and the pencil descends towards *D*. The diagram, as here indicated, shows that the exhaust-port is opened too late, for this corner of the diagram should be very nearly square (see dotted line *p*). The engine

piston now commences its return stroke, and the line *gh* is traced, representing the exhaust-line; and, before reaching the end of its stroke, it commences to rise again at *h*, thus indicating that there is some pressure arising from the compression of the steam and vapor remaining in the cylinder. This is due to the closing of the exhaust-port at *h* before the end of the stroke, causing the curved line *hi*. The rounded corner at *B* shows that the valve is wanting in "lead:" or, in other words, the steam-port was opened too late, as is also the case at the exhaust-end; in the latter case, showing that the release of the exhaust steam is not early enough, and that in consequence of this the back-pressure at the commencement of the return stroke is much too high. This shows that the slide-valve was improperly set,—a defect which can be remedied by shifting the eccentric slightly ahead. This will improve the exhaust, by causing an earlier opening, shown by the dotted curved line *g;* also causing earlier compression, as shown by the dotted line at the point of compression, as well as

the increased lead and initial steam pressure at *B*. The power exerted is thus increased at least ten per cent with the same amount of steam. The steam-line should be parallel with the atmospheric line up to point of cut-off, or nearly so. Should it fall, as the piston advances, the opening for the admission of steam is insufficient, and the steam is *wire-drawn.*

The point of cut-off on all engines should be sharp and well defined; if otherwise, it shows that the valve does not close quick enough.

By having an indicator at each end of the engine cylinder, the back-and-forth action of the steam in the cylinder is simultaneously recorded in the form of a diagram, as before stated, by horizontal and vertical lines and curves. This diagram comprises time of admission, steam-line, point of cut-off, expansion curve, terminal pressure, point of exhaust (or relief exhaust) line, back-pressure line, compression curve, initial pressure, and initial expansion. From these records the total work done by the steam can be accurately ascertained. Very

accurate measurements have been made by the indicator; but the average area of indicator cylinders is only about one-half of a square inch, while that of cylinders indicated may vary from ten square inches to as many square feet. By the use of the indicator, the determination between *nominal* (calculated), *indicated* (real), and *effective* horse-power is found, — the variations between which are very marked.

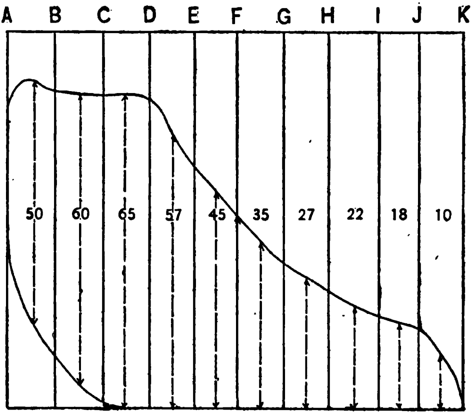
The indicator also furnishes one of the data for ascertaining the power exerted by the steam-engine; namely, the *mean* or *average* pressure of the steam during the stroke on each square inch of the piston. Stated more accurately, it shows the excess of pressure on the steam side of the piston to produce motion, over that on the exhaust side to resist it; and from no other source can it be so accurately ascertained.

The pressure in the boiler is readily known; but the steam in its passage to the cylinder is subject to various losses, such as wire-drawing, condensation, friction, etc., so that frequently the pressure on the piston does not exceed two-thirds of that on the boiler.

HOW TO ASCERTAIN THE HORSE-POWER OF A DIAGRAM.

A horse-power being 33,000 pounds raised one foot high per minute, to ascertain what an engine is exerting all we need

Fig. 12.



SCALE 1" = 40 POUNDS

is to find out how many pounds weight it will raise in a minute, and through how many feet. The above card we will take as an example.

This diagram has registered on it all

the lines and pressures heretofore mentioned.

The connection between this curved figure, and the power exerted by the engine, is not apparent at first sight; and, before showing how to obtain it, it is necessary for the reader to look back to the beginning of this article, to see what is the true measure of power exerted.

Without a clear and definite conception of what constitutes mechanical work, — or, in other words, what is the measure of work done, — it is impossible to form any idea, either of what is meant by economical use of steam, or of the connection between the indicator diagram and the indicated horse-power.

The simplest example of an expenditure of power, and also the most common, is that of a weight raised from the ground. If one pound has been raised one foot high, just half the work has been required which would be required to raise two pounds one foot high. This is so simple a conception as not to require further explanation. A little consideration will show that, generally

speaking, the work required to lift any weight to any height may be said to be equal to a certain number of pounds raised one foot high; or, as it is generally stated for the sake of shortness, the work expended is equal to a certain number of foot-pounds. One pound raised one foot is one foot-pound.

It is a well-known law in mechanics, that when work is done, or power expended, some resistance has been overcome through some distance. What is really done in raising a weight is to overcome the attraction of the earth, or gravity. We make gravity a general standard of resistance, and whenever any resistance is overcome we may refer it to this standard. One pound raised one foot high may be taken as our standard unit of work done.

An engine at work overcomes some resistance, either propelling itself, a vessel, pulling a train, or driving machinery; and the amount of work expended by the engine in overcoming this resistance through a certain distance is equivalent to a certain number of pounds raised through a certain number of feet.

WHAT AN INDICATOR DIAGRAM SHOWS.

Let me refer to the indicator diagram, Fig. 12, as it came from the indicator. This diagram is the result of two movements; namely, a vertical movement of the pencil proportioned to the steam pressure acting on the piston of the indicator, and a horizontal movement of the paper simultaneous with the stroke of the engine piston. It consequently represents, in its length, the stroke of the engine on a reduced scale; and by its height, at any given point, the pressure on the engine piston at a corresponding point in the stroke. The pressure shown is measured by a scale marked to correspond with the indicator spring used. The most common scales are 20, 30, 40, and 60 pounds per inch; that is, an inch of vertical height on the diagram represents 20, 30, 40, and 60 pounds of steam per square inch on the piston, according to the scale of the spring used. These scales are sent with every instrument.

The length of the diagram, measured horizontally, represents the space passed

through by the piston in feet, and is known when we are given the length of the engine crank.

As the pressure is given by the indicator scale in pounds, the area of the figure represents the work in foot-pounds done by the steam on *one* side of the piston in one complete stroke, or a revolution of the crank.

The diagram Fig. 12 is divided into ten equal spaces. The distance from *A* to *B* is one-tenth of the whole length of the indicator card; and, during the time the drum moved horizontally from *A* to *B*, the piston of the engine travelled one-tenth of its stroke; while the drum travelled from *B* to *C*, the piston of the engine moved another tenth of its stroke; and, when the engine had travelled its whole stroke, the drum would have travelled from *A* to *K*, and similarly on the return stroke. It is not a matter of any importance what the length, *AK*, is when compared with the stroke of the engine: so, for convenience, *AK* is usually made about four inches. Excepting in engines running at high speeds, the length is reduced as much as possible

compatible with accuracy. This is to avoid errors caused by shock and jar of changing direction of motion of the paper-drum of the indicator. All that we care about is, that, when the engine piston has moved through a portion of its stroke, the drum shall have moved with it in reduced proportions; and that the motions of both correspond at all points throughout the stroke. Then we have only to look at the indicator card to see what pressure of steam there was in the cylinder at any part of the stroke. In this particular case a $\frac{1}{40}$ spring was in the indicator; which means, that, for every one pound pressure on the square inch of the indicator piston, the pencil of the indicator will rise $\frac{1}{40}$ of an inch.

If we have 80 pounds cylinder pressure, the pencil will rise two inches as soon as the steam is admitted at line *A;* then, as the engine and card move, the pencil moves to *B*, then to *C*. Between this point and *D*, the steam is cut off; then the steam pressure falls as the piston moves on, and the pressure can no longer compress the spring the full height: the pencil falls to *F*,

then to G, and so on to K, where the steam is exhausted into the air; and, the spring being no longer compressed, the pencil falls to the line called the atmospheric line, — a line of no pressure, which will be referred to hereafter.

At the bottom of line K the piston begins the return stroke, and up to C the steam continues to exhaust into the air. At about this point the slide-valve closes; and what is left in the cylinder is compressed until nearly the top of line A is reached, when the slide-valve again opens, and steam is admitted for the next stroke, and the spring compressed as before to the top of line A.

From the curved figure thus formed we find what power the engine was developing. To calculate this area approximately, it is usual for engineers to measure the pressure at ten places, equally distant from each other, on lines drawn between the lines A, B, etc. (see dotted lines, Fig. 12). These pressures added, and divided by ten (the number of spaces), will give the *mean effective indicated pressure* acting on the piston during one stroke.

To find the foot-pounds raised per minute, we multiply the area of piston by the mean pressure, by the revolutions per minute, and by the stroke multiplied by two.

To find the horse-power, we divide the foot-pounds by 33,000. This quotient is the indicated horse-power of the engine; or

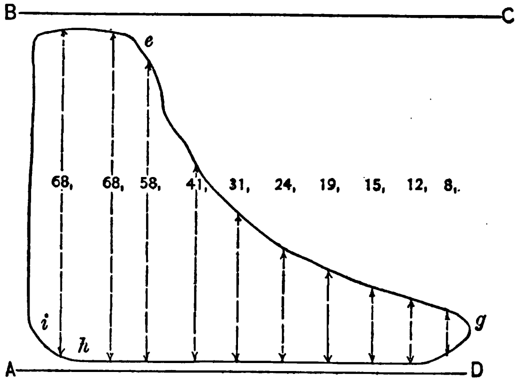
$$\frac{\text{Area of piston} \times \text{mean press.} \times \text{rev.} \times \text{stroke} \times 2}{33000}$$

Where there are a number of cards taken from the same engine to be calculated, we find what is called "the constant" for the engine. This constant is the horse-power which would be exerted by one pound of mean pressure, and is found by multiplying the area of the piston by the feet travelled by the piston per minute; that is, multiplying the area of the piston by the revolutions, by the stroke, and by two, and dividing the product by 33,000. We find the horse-power of this particular engine by multiplying this constant by the mean pressure.

In illustration of the rules given before, we will compute the horse-power exerted in the following diagram, taken from the

cylinder of a Corliss engine. The diameter of piston was 6 inches, the length of stroke 16 inches, and the revolutions per minute 108; diameter of piston-rod, 1½ inches.

Fig. 13.



What is the horse-power of this engine by the indicator?

Cylinder, 6 inches; stroke, 16 inches; revolutions, 108; boiler pressure, 70 pounds; To find the mean effective pressure on the piston, proceed as follows: —

Divide the card into ten equal parts, and measure the length of each ordinate by the

scale corresponding to the spring of the indicator (which, in this case, was 40 pounds for each inch in height). The sum of the length of the ten ordinates amounts to 344 pounds, which, divided by the number of ordinates (ten in this case), gives an average mean effective pressure of 34.4 pounds.

A horse-power is a conventional term, and expresses a rate of mechanical work, measured in foot-pounds for some unit of time, as one second, or one minute: 550 pounds raised one foot high in one second, or 33,000 pounds in one minute, is commonly understood to mean a horse-power.

To calculate the indicated horse-power, multiply the area of the piston in square inches by twice the length of stroke in feet, and the products by the number of revolutions per minute. (This product is known as the "piston displacement.") Divide this product by 33.000, and the result is the "horse-power constant," or the power developed for every pound of mean effective pressure. Multiply the quotient by the mean effective pressure (ascertained from the diagram), and the result will be the indicated horse-power.

Area of piston $= 6 \times 6 \times 0.7854$

$= 28.274$ square inches.

Area of piston-rod $= \dfrac{1.5 \times 1.5 \times 0.7854}{2} = 0.883.$

Average area of piston less one-half area of rod equal 27.391 ($28.274 - 0.883 = 27.391$).

Speed of piston in feet per minute

equal $\dfrac{16'' \times 2 \times 108}{12} = 288'.$

The constant for this engine is, therefore,

$$HP = \frac{27.391 \times 288}{33000} = 0.239 \text{ horse-power constant.}$$

The mean pressure, as per diagram, is 34.4 pounds, and the power developed was

$$HP = 34.4 \times 0.239 = 8.22 \text{ horse-power.}$$

Where great accuracy is required in estimating the power of steam-engines from indicator diagrams, care should be taken to calculate the power of forward and back strokes separately, as the mean effective pressures are not always alike.

In this manner the power exerted by an engine may be ascertained under every variety of circumstances, and also the power required for every kind of machine.

Measuring the power required by a single

machine, among many running in a manufactory, requires great care, but can be done with certainty, even to a small fraction of a horse-power. It is necessary that every thing should be in the same condition during the whole experiment. The proper time to test is after running for several hours, and directly after stopping, when every thing is in the best working condition: say at noon-time. Then, first indicate for the shafting alone; afterwards put on the machine to be tested, — the power required for which is to be ascertained after it has been running for a few minutes; and, finally, after the belt has been thrown off, indicate for the shafting again.

In case the pencil should run over the paper several times, it should be ascertained if it follows the diagram exactly when removed a little from the paper. The first and third diagrams (that is, the friction diagram of the shafting) should be identical, and the excess of the second diagram is the power required by the machinery tested. Care should be observed that all the diagrams are taken at the same speed of the engine.

In all cases the greatest pains should be taken to determine if the diagrams are a true representation of the power exerted: see if the pencil will repeat the diagram, both when in contact and when not in contact with the paper. Often the diagram will not repeat exactly. Whenever this is the case, the pencil must be allowed to run over the paper a sufficient number of times; and the average of all the figures must be taken as the true one.

As before stated, the indicator card is usually run out — or, in other words, the mean pressure of the card is usually ascertained — by reading off, with the aid of the scale, the different mean pressures on each of the ten spaces, then adding them together, and dividing them by ten, or whatever number of spaces there are. This is correct, provided each reading is an accurate one. The following is a far better and easier method: Take a long strip of paper, say one-half an inch wide, and from 10 to 20 inches long, according to the nature of the card; mark a starting-point on the edge near one end; then lay the strip

of paper along the first dotted line, and mark off the length of that line; then lay it on the second space, and add the length to the second dotted line; and so on to the tenth dotted line. By this means, the lengths of all the ten lines are laid end to end. If we now take a rule, and read off how many inches there are in the whole length, and divide them by ten, we get the number of inches in the mean pressure of the whole card. Generally expressed, we multiply the total number of inches read off the strip, by the scale, and divide by ten. This is one of the best and safest ways, if not the very best way, of finding the mean pressure of a card. It is certainly greatly superior to the method of reading off ten different pressures, and adding them together, and dividing by ten, as heretofore described.

There is another method of measuring cards, which at once disposes of many vexatious causes of error. It is by a most ingenious instrument called a planimeter, which is now mostly used for finding mean pressures. This instrument is about as

large and as complicated as a pair of dividers. It will give the area of any card, however awkward in shape, in one minute, by passing one leg of the instrument along the outline. It gives at once the mean effective area, without any second measurement being required for counter-pressure; or it measures any of the areas of a card which may be desired. No skill or mathematical knowledge whatever is required to use the instrument.

One leg of the instrument is caused to remain stationary, and a tracer on the other leg is passed along the outline in one direction till it returns to the starting-point. The readings taken from a counter on the instrument give the area of the enclosed figure. Marvellous accuracy and perfect simplicity are the marked features of the planimeter.

In working out a number of cards with a planimeter, it is most important to remember that the length of the card must be taken into account, because this generally varies to a slight extent in cards taken from the same engine with the indicator;

and it does not do to assume a common length for them all.

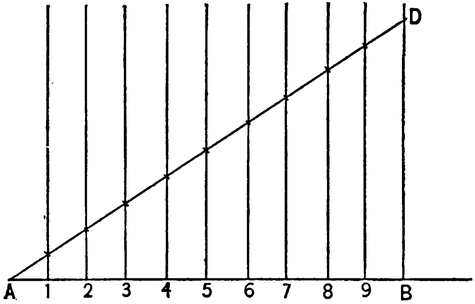
HOW TO DIVIDE A LINE INTO A REQUIRED NUMBER OF EQUAL SPACES.

A foot-rule or scale is usually divided into inches, halves, quarters, and eighths of an inch; and, when the line to be divided into a required number of equal spaces is a multiple of those spaces in any of three units of measurement, it is, of course, easy so to divide it. Thus it is easy, by applying the rule, to divide a line four inches long into four inch spaces, or eight half-inch spaces, or sixteen quarter-inch spaces, or thirty-two eighths-of-an-inch spaces. But, when the line is not such a multiple of the space, it cannot be divided by applying the rule to it; and the following method may be used: —

A line $4\frac{3}{8}$ inches long, to be divided into 10 equal spaces. First draw a line at right angles to the given line, at one end of it; then take a strip of paper, and, applying the rule to the strip, mark off on it 10 equal spaces, which together will exceed

the length of the given line; then place one end of the strip at the open end of the given line, and carry the other end of the strip up until the last point marked off on it touches the right-angled line, and through the points on the strip draw lines

Fig. 14.



parallel with the right-angled line to the given line; and the given line will be divided as required.

Thus, let AB (Fig. 14) be the given line; draw BD at right angles to it; the first 10 equal spaces on the rule which will exceed the length of AB ($2\frac{1}{16}$ or 2.0625) will be 10 one-quarter inches; mark these 10 one-

quarter inches off on a strip, *A* to *D;* place the end *A* of the strip to the end *A* of the line, and move up the strip until the point *D* touches *BD;* and, through points on the strip, draw lines 1, 2, 3, 4, 5, 6, 7, 8, and 9, parallel with *BD;* and the line will be divided into ten equal spaces.

To those who do not have a planimeter, and are frequently in the habit of computing the horse-power of engines from indicator diagrams, this method will be found very advantageous.

A diagram from a condensing, or "low-pressure," engine differs from one produced by a non-condensing, or "high-pressure," engine; from the fact, that, in the former, the line of back-pressure, instead of being a little above atmospheric pressure, approaches more or less to that of perfect vacuum.

In calculating the power from diagrams of condensing, or "low-pressure," engines, it is usual to measure the area above and below the atmospheric line separately. This method gives the value of the average vacuum obtained, and thus indicates the extent

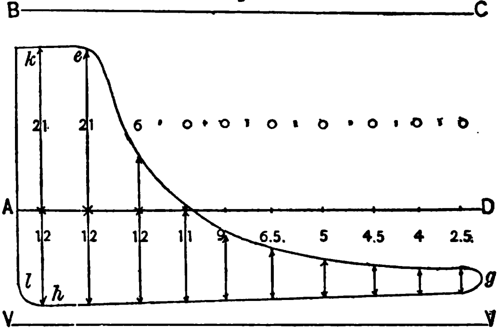
to which the back-pressure is reduced below atmospheric pressure. (See diagram Fig. 15.)

In this the average mean pressure due to the steam was

$$21 + 21 + 6 = 48 \text{ lbs.},$$

which, divided by 10 (the number of divis-

Fig. 15.



ions on the card), equals 4.8 pounds; and the average vacuum realized was

$$12+12+12+11+9+6.5+5+4.5+4+2.5=78.5 \text{ lbs.},$$

which, divided by 10, equals 7.85 pounds: showing that the power realized in this case by removing the resistance of atmosphere

was about *sixty per cent* of that shown by the indicator ; thus, —

$$\frac{7.85 - 4.8}{4.8} = 60 \text{ per cent.}$$

In well-constructed engines, with an early cut-off, the expansion curve *eg* (diagram Fig. 15) will often cross the atmospheric line *AD* before the piston has moved half the length of the cylinder. In such cases as this, the mean pressure represented by the area above the atmospheric line *AD* will be less than below it, which difference is due to the reduced back-pressure by reason of the comparative vacuum in the condenser. The above diagram, Fig. 15, indicates a large amount of expansion.

INDICATED HORSE-POWER.

The *indicated horse-power* is the power developed by the steam on the piston of the engine, without any deduction for friction. The indicated horse-power is calculated from the diagram or cards taken by the application of the indicator to the steam-engine cylinder. It is the total unbalanced power of an engine employed in overcoming

the combined resistance of friction and the load.

EFFECTIVE HORSE-POWER.

The *effective horse-power* is the actual and available horse-power delivered to the belt or gearing, and is always less than the indicated, from the fact that the engine itself absorbs power by the friction of its moving parts.

ENGINE FRICTION.

The power absorbed in driving an engine against its own friction is a most variable quantity. With a good and well-constructed engine, having ample bearing surfaces, efficient means of lubricating them, and valves nearly balanced without over-complication, the friction may not exceed *ten per cent* of the indicated power; but in badly constructed engines the friction may be nearer fifty per cent. In the case of an engine having ordinary unbalanced slide-valves, it is probable that quite one-third of the whole frictional resistance is due to the valve. The heat due to the internal engine friction

— that is to say, the friction of the valves and piston — is imparted to the steam; and either the whole or greater part of it is carried to the condenser or atmosphere with the exhaust steam.

The power absorbed in overcoming friction is not only wasted, but it is wasted in wearing out the engine.

In the diagram, Fig. 13, the calculation gave what is called the indicated power; that is, the effective available power of the engine. It does not show the gross or whole power of the engine. This gross power is reduced to effective motive power in three ways; namely: —

First. In expelling the steam left in the cylinder at the end of the stroke; the expelled steam carrying its *heat* with it to the atmosphere in a non-condensing, or "high-pressure," engine, and to the condenser in a condensing, or "low-pressure," engine.

Second, In compressing the steam in the cylinder after the exhaust-port is closed; but, as this steam is again used after compression, the power used in compressing it is not necessarily wholly wasted.

Third, In overcoming the friction of the moving parts of the machinery; including, in locomotives, the friction on rails, and, in stationary engines, the friction of the belt or gearing.

The effective available motive power will therefore vary in proportion to the power lost through these reducing causes. The less power required to expel and compress the steam left in the cylinder, and to overcome the friction, the greater will be the effective motive power; and *vice versa*.

In calculating this power, however, from a diagram, only the first and second of these causes are or can be considered.

The piston of an engine is always acted upon by two opposing forces,— one propelling and the other repelling; and the difference between them is what, in practice, is called the effective motive force or power.

The propelling force must, of course, in all cases, be sufficient at least to overcome the repelling force, or back-pressure. This back-pressure, as will presently be seen, is always greater in non-condensing, or "high-pressure," engines than in condensing, or

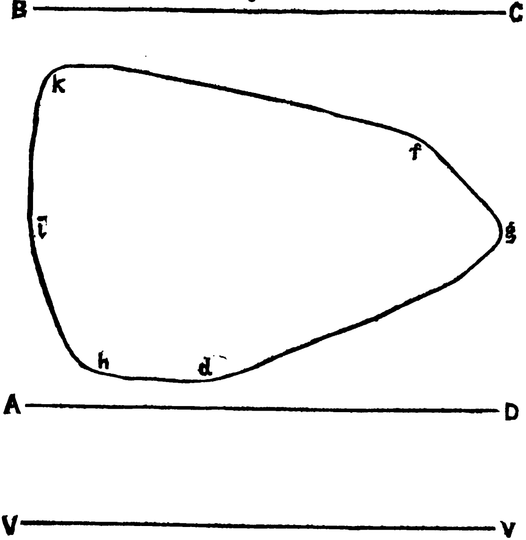
"low-pressure," engines. In the former, the propelling steam left in the cylinder at the end of the stroke — that is, the exhaust steam — escapes, or is expelled into the air: in the latter, into the condenser. In the former, the back-pressure must necessarily be at least the pressure of the atmosphere, which averages about 14 pounds to the square inch; but it is always greater than this, because of the friction of the exhaust steam in the ports and pipe connections; and in badly constructed engines it is much greater. In condensing, or "low-pressure," engines, the back-pressure should always be less than the pressure of the atmosphere, depending upon the approximation to vacuum obtained in the condenser.

In the diagram, Fig. 16, taken from a non-condensing engine, it will be seen that the back-pressure line *gdh* is considerably above the atmospheric line *AD*, and indicating excessive back-pressure.

Excessive back-pressure in a non-condensing engine is caused by or results from too great impediment to the escape of the exhaust steam; and, in condensing engines,

to imperfect vacuum in the condenser. The value of the indicator in revealing defects of this kind cannot be over-estimated.

Fig. 16.



The difference between a non-condensing and a condensing engine is, as has been seen, that in the former the exhaust steam escapes or is expelled, more or less directly,

according to the construction of the port-passages and pipe connections, into the air; and, in the latter, into the condenser.

In the former, the back-pressure is the pressure of the atmosphere, increased more or less as the escape of the exhaust steam is more or less impeded. In the latter, the back-pressure depends chiefly upon the pressure of the exhaust steam, or, in other words, the degree of vacuum in the condenser.

The condenser is an air-tight iron box or vessel, fitted with valves, and connected more or less directly by pipes with the exhaust-ports of the cylinder. Its office is to condense the exhaust steam, and thus prevent the back-pressure of the atmosphere. The condensation of the steam is usually effected either by bringing it into contact with a jet of cold water, or by passing it through or about a series of tubes on the other side of which cold water is circulated. In either case the steam is condensed almost instantly; and a vacuum, more or less perfect, is formed in the condenser. A perfect vacuum cannot, in practice, be had; but an

average of about 26 inches, or 13 pounds, is usually obtained by the gage. Diagrams generally show from 3 to 4 pounds less. The approximation to a vacuum, and corresponding diminution of back-pressure, are affected in three ways; namely, —

First, The temperature of the condensing water.

Second, The pressure of the atmosphere.

Third, The friction of the exhaust pipes and ports.

First, If the temperature should be 32° Fah., the pressure would be only 0.085 pounds to the square inch, and the vacuum as nearly perfect as is obtainable. The condensing water is, however, usually taken at 40° to 80°, and leaves the condenser at from 90° to 120°; making the temperature in the condenser generally about 100°, which would give a back-pressure from this cause alone, of about one pound to the square inch.

Second, If the barometer stands at only 28 inches, 13.7 pounds would be a perfect vacuum, 30 inches of mercury being equivalent to 14.7 pounds; and, if the water in

the condenser be at a temperature of 130°, its vapor will form a resistance of 2.21 pounds: therefore, the lowest attainable vacuum would be but $13.7 - 2.21 = 11.49$ pounds. Whereas, if the barometer stood at 31 inches, a perfect vacuum would be 15.2; and, if the water was but 100°, its vapor would give a resistance of only 0.9 pounds: and consequently the highest attainable vacuum would be $15.2 - 0.9 = 14.3$ pounds, making a difference of 2.81, or a gain of *twenty per cent.*

Third, The friction of the exhaust pipe and ports will be excessive, if they are too small, to the same extent as in the case of non-condensing engines.

The water used for steam-engine purposes invariably contains more or less air, which, if allowed to accumulate, would gradually destroy the required vacuum. It is necessary, therefore, to draw off this air as well as the water, and this is done by means of an "air-pump" worked by the engine; and, of course, the power required to do this, although needfully expended, is so much power to be deducted from the

given power, reducing the efficient motive power of the engine. The power thus expended is usually equivalent to from one-half to one pound pressure. But it is frequently necessary to raise the condensing water from a lower level to the line of the condenser; and in that case the power required to do this work is also power to be deducted from the gross power, also reducing the efficient motive power of the engine. In all cases it is only the net motive power, after deducting the power needed to overcome the back-pressure, that is represented in the area of the diagram.

The pressure of the atmosphere is usually taken as 15 pounds, which is too high, being correct only when the barometer stands at 30.54 inches, — a most unusual occurrence. But the error is unimportant, and it is very convenient to avoid the use of a fraction.

The principal object of knowing the exact pressure of the atmosphere is to ascertain the duty performed by the condenser and the air-pump. The temperature of discharge being known, the pressure of vapor inseparable from that temperature is also

known (see Nystrom's "Pocket Book," p. 400) ; and, this being deducted from the actual pressure of the atmosphere, the remainder is the vacuum in which the water would boil. The power of the air-pump is shown in the closeness with which the vacuum approaches this point.

The vacuum shown by the indicator will generally vary from that shown by the vacuum gage when it is constructed with a glass tube hermetically sealed at the top; for such gages are designed to show the variation from a perfect vacuum without reference to the weight of the atmosphere: but the vacuum shown by an indicator is affected by all its variations.

AN IDEAL DIAGRAM SHOWING THE ACTION OF STEAM.

Some of the disturbing causes on diagrams of steam-engines, which make the real differ from the ideal form of the diagram, have already been considered incidentally. At present, the more important and usual of these deviations are to be classed and considered in detail.

These causes affect the power of the engine, as well as the character and shape of the diagram.

The indicator diagram is, of course, the key to the action of the steam in the cylinder. A part of the work performed by the steam is spent in overcoming the friction of the engine itself; and, consequently, the efficiency of the *engine* is most fairly tested by the amount of external work absolutely performed against a brake, or otherwise.

Where the efficiency of the *steam* alone is concerned, however, the diagram is the only true criterion; and it will be necessary to deal with its theory carefully to prevent misunderstandings, which are frequent in practice.

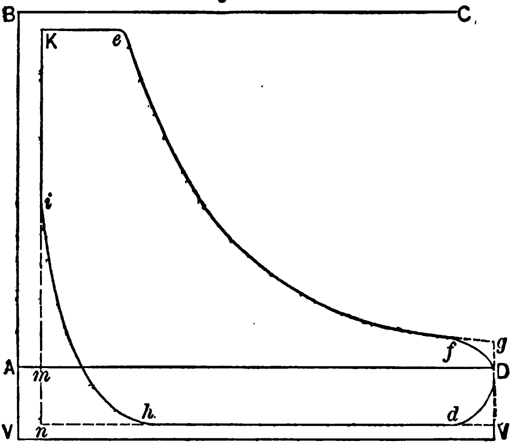
THE ACTION OF STEAM IN THE CYLINDER.

The action of steam in any steam-engine cylinder is best understood from a diagram representing the varying pressures and volumes through the stroke.

Such a diagram is usually obtained by an indicator applied to the cylinder, and in such case the pressures shown are actually

those of the steam in use. For purposes of comparison and calculation, however, it is more convenient to construct an ideal diagram, as nearly as possible such as would

Fig. 17.



be given by an indicator applied to an engine as nearly perfect as practicable, working under the same conditions. Such a diagram is shown at Fig. 17, where horizontal distances represent volume, and vertical distance pressure.

The several lines on the ideal diagram will be designated here, reference being had to this diagram.

The base-lines of the theoretical diagrams are as follows: —

THE ATMOSPHERIC LINE.

When the atmosphere has free access to both sides of the piston of the indicator before steam is admitted, a straight line, AD, will be drawn by applying the pencil to the moving paper. This line is called the line of atmospheric pressure, or zero, on the steam-gage. From this line we measure pressure for non-condensing, or "high-pressure," engines.

THE LINE OF PERFECT VACUUM.

The line VV represents it. This line cannot be drawn by the indicator, but must be drawn by hand, parallel with the atmospheric line, and at the proper distance below it, to represent the pressure of the atmosphere as shown by the barometer, according to the scale of the indicator diagram. When the actual pressure is not

known, it is to be assumed at 14.7 pounds on the square inch, corresponding almost exactly with 30 inches of mercury, which is about the average pressure at the level of the sea. The barometric column falls $\frac{1}{100}$ of its height for every 262 feet of elevation above the sea-level.

THE LINE OF BOILER PRESSURE.

This line is represented by the letters *BC*, and is also drawn by hand, parallel with the atmospheric line, and at the proper distance above it, to indicate the steam pressure per square inch, as shown by a correct steam-gage, measured off by the scale of the indicator diagram. It can be drawn by the indicator attached to the cylinder, only when the engine is at rest, and while an equilibrium of pressure is established between the boiler and cylinder. It is generally somewhat higher than the initial pressure in the cylinder.

THE CLEARANCE LINE.

This line is represented by *BV*, and is at right angles to the atmospheric line *AD*,

and at such distance from *kimn* that the included space, *BAV* and *nmik*, correctly represents the clearance.

This clearance is the cubical contents of the steam-port passages and the space between the piston and the end of the cylinder, or head, to which it is nearest at the end or beginning of a stroke. Supposing them, when added together, to be at each end one-twelfth of the whole cubical contents of the cylinder for one stroke of the piston, then the distance *Am* would be made one-twelfth ($\frac{1}{12}$) of *mD*. In the diagram, Fig. 16, one-twentieth ($\frac{1}{20}$) has been taken, so that the line *Am* is one-twentieth ($\frac{1}{20}$) of the length of *mD*. It is necessary to take these cubical contents into account, for the passages and clearance must always be filled with steam at each stroke, which is compressed and expands just precisely the same as the rest of the steam in the cylinder does after the steam has been cut off. It is necessary to draw this line, and to add this space to the indicator diagram, whenever the theoretical curve is constructed to compare with the actual curve traced by

the indicator, and must be reckoned as part of the diagram in calculating the average pressure, and in producing the theoretic curve, or line of perfect expansion. The clearance is, however, rarely given; and it varies in different engines from 1 to 20 per cent of the space swept through by the piston in one stroke. If we have the drawings of the engine, we can calculate it; if we know the style of engine, we can approximate it.

The best method, providing the piston is tight, is as follows: —

Put the engine on the centre, remove the valve-chest cover, uncover the steam-port on the end where the piston is, fill the steam passage and piston clearance with water, level with the valve-seat; allow it to remain a few minutes, and, if it maintains its level, it is evident the piston is tight: then draw off the water, measure or weigh it, reduce it to cubic inches, and we have it exactly. The number of cubic inches of clearance, divided by the cubic inches of space swept through by the piston in one stroke, gives the ratio of cylinder capacity

to clearance. This matter will be more fully illustrated hereafter.

DIVISION OF THE OUTLINE DRAWN BY THE INSTRUMENT DURING A REVOLUTION OF THE ENGINE.

The diagram, Fig. 17, shows all the lines that would be traced by the pencil of the indicator during one revolution of the engine, assuming the action of the steam to be nearly theoretically correct. In order that the student may better understand the subject-matter, the following names have been given to the lines represented, as follows: —

The line from *i* to *k*, the admission line.
The line from *k* to *e*, the steam line.
The line from *e* to *f*, the expansion line.
The line from *f* to *d*, the exhaust line.
The line from *d* to *h*, the back-pressure, or line of counter-pressure.
The line from *h* to *i*, the compression and lead line.

Of these divisions the first four are drawn during the forward stroke of the piston, and

until it is at, or very close to, the termination of its stroke; and the last two are drawn during the return stroke.

ADMISSION LINE.

The admission line *ik* shows the rise of pressure due to the admission of steam to the cylinder. This line is generally very nearly vertical; and, when this is the case, it shows that steam of nearly boiler pressure is had at the commencement of the stroke, while the piston is nearly stationary. Should this line incline forward, as shown at *B* in Fig. 11, or, as at *k* in Fig. 16, curve with the steam line, the reverse is indicated; or should this line continue vertically beyond, and then suddenly drop to the level of the steam line, it signifies that the steam is wire-drawn, and cannot keep up the full pressure as the piston starts forward: but should this line, after projecting above, be suddenly depressed below the level of the steam line, vibrating back and forth one or more times on the latter line with acute angles of return, it may be attributed to the momentum of the reciprocating parts of the

indicator while running at very high speeds. This will be hereafter more fully explained.

THE STEAM LINE.

The steam line *ke* is traced while the steam is being admitted to the cylinder, and should be nearly parallel to *BC*, and is invariably several pounds pressure below it; this loss in pressure occurs from radiation and friction in the pipes from the boiler to the cylinder. This line also represents the initial pressure acting on the piston up to the point of cut-off, and should be of unvarying height, to show that full boiler pressure is maintained. It also shows, at its termination, the point at which the valve closes, or steam is cut off.

To maintain a proper steam pressure in the cylinder, depends, of course, in the first place, upon the amount of steam-port area. It will be noticed in diagram Fig. 10, taken from a Corliss engine, that the piston obtained nearly the full boiler pressure at the very commencement of the stroke. The initial cylinder pressure was 97 per cent of the pressure in the boiler; while in the

diagram Fig. 16 (fitted with the ordinary slide-valve, and the steam controlled or regulated by a valve in the steam-pipe), the maximum cylinder pressure reached but 88 per cent of the boiler pressure, notwithstanding the slower speed of the engine, — the former making 90 and the latter but 40 revolutions per minute.

An important consideration in connection with the admission of steam is that the maximum cylinder pressure be fully maintained until the closing of the valve; in other words, that the steam-line traced by the indicator should, as much as possible, run in a horizontal direction. (See diagrams, Figs. 10, 12, 15, and 17.) To effect this it is necessary to have the steam-port fully uncovered early in the stroke, so that the steam can be rapidly introduced into the cylinder. Referring to the above-mentioned diagrams, we find that the steam-line is kept well up to the boiler pressure, and this pressure is nearly fully maintained until the point of cut-off is reached. If we take into consideration the small amount of lead obtained in these cases, we must at-

tribute the comparative good results solely to the employment of Corliss and Buckeye valves, which permit, with a smaller amount of angular advance of the eccentric, a very rapid and good introduction of steam.

In locomotive engines the diagrams taken with a high rate of expansion, more particularly at high speeds, the steam-line generally falls more or less during the period of admission, indicating that the steam-port opening is too small.

THE POINT OF CUT-OFF.

This takes place at *e*. In the theoretical diagram the corner is abrupt, but in practice it is more or less rounded. The diagram does not always show clearly the exact point where the convex curve of the rounded corner changes to the concave curve of the expansion line; but the point of cut-off is properly located at the point where the direction of curvature changes from convex to concave.

THE EXPANSION CURVE.

This is represented by the line *efg*, and

results from a fall of pressure due to the expansion of the steam remaining in the cylinder after cut-off takes place. The actual curve, as drawn by the indicator, will be above the theoretical curve laid down by the law of Boyle, and Mariotte's law: that is to say, the pressure is inversely as the volume; and the curve which expresses the pressure for every point of the stroke is an equilateral hyperbole. In all indicator diagrams a material difference will be noticed between the true ratio of expansion and the corresponding pressures; the amount of departure of the actual pressures from the theoretical curve bearing, however, a certain relation to the degree of expansion, as will be seen hereafter.

There are various causes which produce this action during the period of expansion, but their precise influence is more or less difficult to ascertain. In the first place, leakage at the valves or past the piston is, of course, calculated to alter the actual expansion curve.

The effect of leakage, if such occurs, is generally easily detected by the irregular

form of the indicator curves. The main cause of the peculiar action of the expanding steam is, according to a large number of experiments made, the heat given off by the cylinder to the contained steam after its communication with the boiler has been cut off. This condition is facilitated by the presence of a certain quantity of water, which, at the commencement of the expansion, has the temperature of the live steam; but, as the pressure is reduced in the cylinder, this water will be instantaneously evaporated, and thus abstract from the cylinder a certain amount of heat. The heat absorbed with such rapidity is sufficient to raise the pressure considerably above that which would have existed had no condensation and re-evaporation taken place. The amount of heat which can be absorbed depends, of course, upon the difference of temperature between the steam and the metal.

On the other hand, the mean temperature of the cylinder is influenced by the amount of protection against radiation and conduction of heat from the cylinder, by the

amount of "throttling" from the boiler to the cylinder, by the extent to which expansion has been carried, and by the speed in revolutions per minute.

When the communication between the boiler and the piston is open, the cylinder will acquire a temperature practically the same as that of the boiler pressure; and, if the cylinder contained nothing but dry or superheated steam, this temperature would probably be maintained for the greater part of the stroke. But owing to a certain amount of water which has been deposited in the cylinder, and which is re-evaporated at the expense of heat imparted to the cylinder, this latter will become materially cooled by the time the piston has reached the end of the stroke.

From these considerations, the relative effect of the various degrees of expansion and of speed will readily be appreciated. As the degree of expansion is increased, the quantity of water converted into steam becomes also greater, — necessitating, however, a larger condensation of high-pressure steam during admission, — and the longer

the duration of the stroke. In other words, the slower the engine is running, the more heat will be absorbed from the cylinder by the conversion of this water into steam.

THE POINT OF RELEASE, OR OPENING, OF THE EXHAUST-PORT.

This is at *f*, diagram Fig. 17. To provide a rapid egress for the exhaust steam, and in order that its pressure may be as nearly as possible at a minimum, after the work in the cylinder has been performed, it is necessary that the exhaust-port should be opened before the piston reaches the end of its stroke. The proper amount of this pre-release depends, of course, upon the velocity of the piston, and the quantity of steam to be discharged, or the grade of expansion. If, on the contrary, the steam be confined until the last instant, the back-pressure at the commencement of the return stroke will be considerably increased, or in proportion to the period of admission. The deficiency of early release produces in the indicator curves a sharp corner at *g* at the end of the stroke, as shown in diagrams,

Figs. 13 and 15. It will be noticed, also, that a considerable loss of effective pressure is caused for the same reason, as clearly shown by the reduction of the area of the indicator diagrams. The amount of back-pressure against the piston during the remainder of the exhaust also depends directly upon the amount of release, and indirectly upon the speed of the engine. If the exhaust-port is not well open at the end of the stroke, it is evident that the greater volume of the steam must be discharged during the return stroke of the piston, until the closing of the exhaust-port; but, as the piston attains its maximum velocity at half stroke, the minimum back-pressure above the atmospheric line must then be greater than it would be under the more favorable condition of premature escape of the steam. Therefore, the non-release of the steam before the end of the stroke involves not only a direct loss of the work done by the steam, as shown by the corner cut off from the indicator diagrams, Figs. 13 and 15, but its injurious effect is also manifest during the greater part of the return stroke.

The loss of work done through an early release of the exhaust is more than regained during the return stroke, the back-pressure against the piston becoming reduced to that of the atmosphere in non-condensing engines. (See diagram Fig. 13.)

THE EXHAUST-LINE.

It is, of course, desirable that the pressure of the steam be got rid of as completely as possible before the piston commences its return stroke. This is accomplished by having the exhaust port and passages sufficiently large, and opening the port a sufficient time before the termination of the stroke, according to the density of the steam to be released, and the velocity of the piston.

The exhaust-line commences at the point of release *f*, Fig. 17, where the expansion curve changes to convex as the pencil travels to the line of counter-pressure, and shows the fall of pressure caused by the release or opening of the exhaust-port for the escape of the steam before the forward stroke is finished, in order to diminish the

back-pressure. In an engine in which there is no pre-release (the exhaust-port opening exactly at the end of the forward stroke), the diagram during the return stroke is usually a curve more or less similar to the line *gd*. (See Fig. 16.)

The lower side of the theoretical diagram, Fig. 17, used in calculations, being the line *VV*, represents the pressure in the condenser; or, in non-condensing, or "high-pressure," engines, the atmospheric pressure line *AD*.

By making the release occur early enough (for example, at the point corresponding to *f*, in diagram Fig. 17), the entire fall of pressure may be made to take place towards the end of the forward stroke, so as to make the back-pressure coincide sensibly with that corresponding to the line *VV*. Then the end of the diagram will assume a figure represented by the line *fDd* in diagram, Fig. 17, which is usually more or less concave. The greatest amount of work is insured by making the release take place at point *f*; so that about one-half of the fall of pressure shall take place at the end of

the forward stroke, from *f* to *D*, and the other half at the commencement of the return stroke, as indicated by the curve *Dd*. The line *fDd* is traced while the excess of pressure remaining at the point of exhaust is being released.

BACK PRESSURE, OR LINE OF COUNTER-PRESSURE.

If the steam used in working engines were unmixed with air, and if it could escape without resistance, and in an inappreciably short time, from the cylinder after having completed the stroke, the back-pressure would be simply, in non-condensing engines (called *high-pressure engines*), the *atmospheric pressure* for the time ; and, in condensing engines, the pressure corresponding to the temperature in the condenser, which may be called the *pressure of condensation*. The mean back-pressure, however, always exceeds the pressure of condensation, and sometimes in a considerable proportion. One reason for this, which operates in condensing engines only, is the presence of air mixed with the steam, which

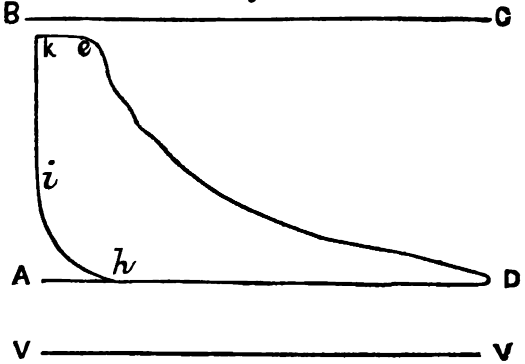
causes the *pressure in the condenser*, and consequently the back-pressure also, to be greater than the pressure of condensation of the steam. For example, an ordinary temperature in a condenser when worked properly is about 100° Fah., to which the corresponding pressure (absolute) of steam is about one pound on the square inch; but the absolute pressure in the best condensers is scarcely ever less than two pounds on the square inch, or nearly *double* the pressure of condensation.

The principal cause, however, of increased back-pressure, is resistance to the escape of the steam from the cylinder, by which, in condensing engines, the mean back-pressure is caused to be from one to three pounds on the square inch greater than the pressure in the condenser.

In non-condensing engines, experiments show that the *excess* of the back-pressure above the atmospheric pressure varies nearly: As the square of the speed. This excess of back-pressure is less, the shorter the cut-off is; in other words, the greater the ratio or grade of expansion: that is

to say, the longer the *time* during which the expansion of the steam lasts. In cylinders with a mean of 16 per cent of release (that is, with the exhaust-port opened when the piston had performed 0.84 of its stroke), with steam cut off at one-half the length of stroke (that is,

Fig. 18.

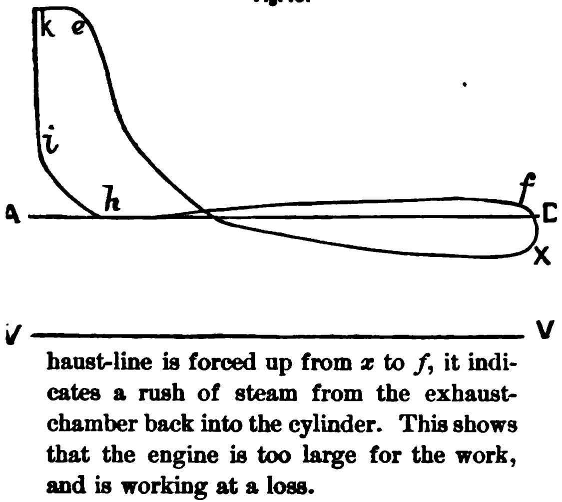


with a ratio or grade of expansion of 2 nearly), and with a piston speed of 600 feet per minute, being the maximum of speed in a good engine, — the excess of the back-pressure above atmospheric pressure was about 0.163 of the excess of the pressure

of the steam at the instant of release above the atmospheric pressure. When the pressure falls during expansion, as in Fig. 18, as low as the return or back pressure, this exhaust-line does not exist.

When the steam is exhausted below the return pressure, as in Fig. 19, and the ex-

Fig. 19.



haust-line is forced up from *x* to *f*, it indicates a rush of steam from the exhaust-chamber back into the cylinder. This shows that the engine is too large for the work, and is working at a loss.

When the steam is exhausted at a high pressure and through cramped passages,

the exhaust-line extends over most of the return stroke, as shown in Fig. 16.

THE BACK-PRESSURE LINE.

This is represented by the line *dh*, and is the pressure behind the piston during the return stroke, and is called back-pressure because it acts in opposition to the return movement of the piston. In diagrams from non-condensing engines (commonly called "high-pressure" engines), it is coincident either with one or more pounds pressure above the atmospheric line (see diagrams, Figs. 13 and 18); while in diagrams from condensing engines (commonly called "low-pressure" engines), it is 22 or 24 inches of vacuum below, or such a distance below the atmospheric line as will coincide with the vacuum attained in the condenser (see diagrams, Figs. 15 and 17). The resistance offered to the escape of the released steam has the effect of reducing, by a corresponding extent, the effective or indicated power of the engine. When the steam escapes from a non-condensing engine, the back-pressure cannot be less than

the atmospheric pressure (14.7 pounds) at the time; and, when it escapes from a condensing engine into a condenser, the back-pressure upon the piston cannot be less than the pressure of vapor existing in the condenser. The excess of resistance over these limits depends chiefly upon the state of the steam, the size and direction of the exhaust-passages, and the speed of the engine. Therefore, the passages and pipes communicating with the atmosphere should be at least *fifty per cent* larger than the ports, and as free from angles as possible.

These requirements apply to condensing engines even more strongly; and, in addition, the condenser and air-pump must be able to maintain a proper vacuum.

THE POINT OF EXHAUST CLOSURE.

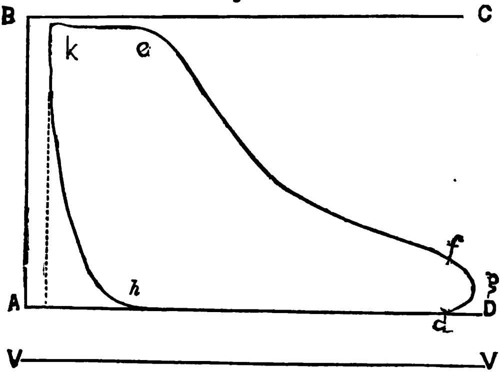
This is represented at *h* in diagram Fig. 17, and is where the exhaust-port is closed against the escaping steam. It cannot be located in all cases very exactly by inspection; for while, like the point of cut-off and exhaust, it is anticipated by a change of

pressure due to a more or less gradual closing of the valve, it is not marked by a change in curvature of the line.

THE LINE OF COMPRESSION, OR CUSHIONING.

This line, when it exists, is formed by closing the exhaust before the end of the

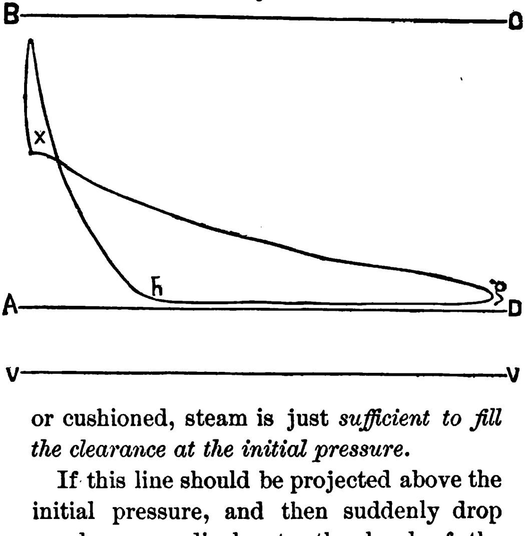
Fig. 20.



return stroke: for example, at the point corresponding to *h* on diagrams, Figs. 16, 19, and 20. A certain quantity of steam in the cylinder is then compressed by the piston during the remainder of the return stroke, and the rise of its pressure is represented

by the curve *hk*. In the reduced diagram, Fig. 20, taken from one of the most advanced types of engines, this curve terminates at *k*, and represents the *most advantageous adjustment* of compression, which takes place when the quantity of confined,

Fig. 21.



or cushioned, steam is just *sufficient to fill the clearance at the initial pressure.*

If this line should be projected above the initial pressure, and then suddenly drop nearly perpendicular to the level of the steam line, thus forming a loop (see Fig.

21), it would indicate an excess of compression, due to closing the exhaust too soon. It is evident that this would be very objectionable, involving a loss of efficiency. In computing such a diagram, the area contained in the loop *x*, at the commencement of the stroke, — denoting negative work, as it were, — should be subtracted from the total area included in the indicator diagram.

Compression also has a useful effect in the working of an engine, by providing an elastic cushion whereby the momentum of the piston and its connections is gradually absorbed, and the direction of motion reversed without "thump" or "shock." There is no "jar" from the entering steam when a new stroke begins. The proper regulation of compression serves to make an engine work easily and smoothly, and consequently reduces the wear and tear of the working parts. The pressure due to the momentum of these parts will, of course, depend upon their weight and velocity increasing directly as the square of the speed. These data being given, the amount of cushion, or pressure, required to counter-

balance work stored up in the reciprocating parts can easily be ascertained. It follows that the compression should decrease rapidly as the speed diminishes, and *vice versa.*

In fast-running engines, especially locomotives, compression also serves to prevent waste from clearance. The capacities of the clearance spaces and the steam-ports are relatively larger than in most other steam-engines, on account of the higher speed of the former. These spaces must be filled, at the commencement of the stroke, with high-pressure steam, which is obtained either by taking a supply of live steam from the boiler, or by compressing into the clearance spaces the low-pressure steam that remains in the cylinder at the closing of the exhaust-port. But in the latter process a certain quantity of steam is saved at the expense of increased back-pressure. It should be borne in mind, also, that the total heat of the compressed steam increases with its pressure; and, as this latter approaches the boiler pressure, the temperature of the steam in compression is also raised from that of about atmospheric

pressure to nearly the temperature of the boiler pressure. These changes of temperature which the steam undergoes will affect the surface of the metal with which the steam is in contact during the period of compression. It follows, of course, that the ends of the cylinder principally comprising the clearance spaces acquire a higher temperature than those parts where only expansion takes place. This is an important consideration, since the fresh steam from the boiler comes first in contact with these spaces; and by touching surfaces which have been thus previously heated by the high temperature of the compressed steam, less heat will be abstracted from the live steam, and therefore a less amount of water be condensed in the cylinder.

Power expended in compression lessens the available power of the engine, without necessarily lessening the efficiency of the steam. Under proper management, as stated above, the compressed steam gives out during its re-expansion the power directly expended in compressing it. There is, no doubt, a somewhat great proportional

loss by friction; but, to counterbalance this, the wasteful back-pressure is reduced by the earlier closing of the exhaust.

The termination of the compression curve should coincide with the beginning of the admission line *ik* (see diagram Fig. 17).

As in expansion, so in compression: the actual curves, as shown by the indicator diagrams generally, and more especially those taken from locomotives, do not coincide with the theoretical curves. Here again the application of the law of Boyle and Mariotte — namely, the volume of the retained steam being inversely as the pressure — comes nearest to practical results. It will not be difficult to account for the fact that the indicated compression curve should be below the theoretical curve. During the period of exhaust, the surface of the cylinder-cover, piston, and cylinder have become materially cooled. When the exhaust-port closes, the pressure and temperature of the retained steam rapidly rise, the temperature of the metal in contact with it rising simultaneously; but, owing to the surfaces being large in proportion to the quantity of steam,

a portion of the steam will be condensed. This loss of compression pressure is attended by a corresponding gain of total useful pressure. Thus the departure of this curve, as well as that of the actual expansion line, below and above the theoretical curves respectively, shows a proportional increase of the power exerted by the engine, which is clearly demonstrated by the increase of area included in the indicator diagrams.

By *lead* is meant the width of the opening of the steam-ports before the beginning of the stroke of the piston. On the steam side of the valve, it is called *outside lead;* on the exhaust, *inside lead.*

The lead and the period of admission should be the same for each end of the cylinder, for each point of cut-off, and if possible, in locomotive engines, in the back as well as the forward gear.

It is found necessary, especially with high speeds of piston, in order to insure good action of the steam, that the maximum cylinder pressure should be attained at the very commencement of the stroke. If the

steam-port is not opened until after the piston has commenced its stroke, especially where there is but little compression, some appreciable time would be consumed in filling the clearance space and the steam passages with steam. In locomotives where the slide-valve is worked by the ordinary link-motion, the steam-port will not open rapidly enough to enable steam of the maximum boiler pressure to fill the space after the receding piston, unless the valve begins to open the steam-port *before* the piston begins its stroke; that is, before the end of its preceding stroke. The Baldwin Locomotive Works allow from $\frac{1}{16}$ (0.0625) to $\frac{3}{16}$ (0.1875) inch lead, according to the class of locomotives; but in ordinary cases, from $\frac{1}{32}$ (0.03125) to $\frac{1}{16}$ (0.0625) of an inch will be sufficient.

When the maximum cylinder pressure is attained at the commencement of the stroke, the admission line of the indicator diagram (the piston being at the end of the stroke) will rise in a vertical line (see diagram Fig. 10); but, if the maximum pressure is not so attained, the admission line will deviate

slightly from the vertical (see diagram Fig. 16).

Lead and compression both regulate the steam admission. If the clearance space at the beginning of the admission is already filled with compressed steam, a less amount of lead is necessary, and *vice versa*.

In locomotive engines with the shifting-link motion, however, not only the lead, but also the compression, increase rapidly as the link approaches mid-gear or half-stroke. This is not a drawback, as the increased compression is calculated to facilitate greatly the attainment of the full pressure of steam in the cylinder at the commencement of the stroke.

Furthermore, it should be remembered that a good admission of the steam depends not only on the amount of lead, but also on the commencement of it; or, in other words, on the period at which the valve opens the connection with the steam-chest preparatory to the next stroke of the piston.

THE MEAN EFFECTIVE PRESSURE.

The mean effective pressure is the differ-

ence between the mean, or average, propelling pressure, and the mean, or average, back-pressure. This pressure is best obtained from indicator diagrams. To arrive at it correctly, we divide the length of the card into ten or more equal spaces, so arranged that there is a half-space at each end (see dotted lines, Fig. 13). Ten is a convenient number, but this is immaterial: any other number may be used. The more numerous the spaces, of course, the greater the accuracy.

THE TERMINAL PRESSURE.

This term is sometimes applied to the pressure at the exhaust-point when the steam is released; but, as it is an indispensable factor in the calculations, it is properly defined as the pressure that would exist at the end of the stroke if the steam had not been released at that earlier point. A continuation of the expansion curve, as at *g* in diagram, Fig. 17 (see dotted line), will explain the method of finding it. Diagrams, Figs. 13, 15, and 16, show that the exhaust has taken place at the end of the stroke:

hence, in those diagrams, terminal and exhaust pressure are the same. This pressure is measured from the extremity of the curve to the vacuum line *VV:* hence it is the *absolute terminal pressure.*

THE INITIAL PRESSURE.

The initial pressure is that pressure which acts upon the piston at the beginning of its stroke up to the point of cut-off; and is always less than that of the boiler, because as soon as the steam leaves the boiler it begins to condense. It can receive no more heat from any source; but it must impart heat to every thing, and supply all loss resulting from radiation. A portion of the steam is always condensed as it enters the cylinder, from coming in contact with the surfaces which have just been cooled down by being exposed to the colder vapor of the exhaust steam. More especially is this so in slow-running engines, where little or no compression takes place.

INITIAL EXPANSION.

Initial expansion is the expansion that

takes place during the admission of steam before the steam is cut off. The steam line *ke*, in diagram Fig. 16, shows considerable initial expansion, which is desirable in a "throttling" engine, — from the fact that *saturated* steam becomes *superheated* during the process of "throttling," — but is not desirable in cut-off engines.

WIRE-DRAWING AND THROTTLING.

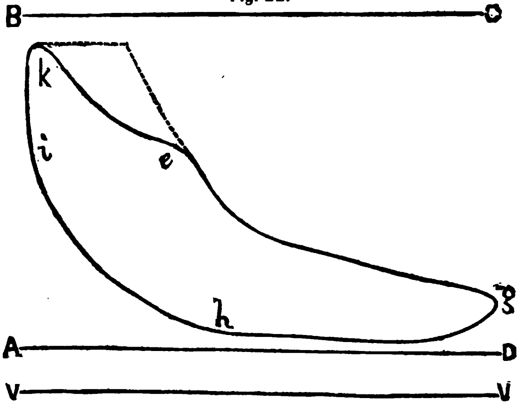
When steam is reduced in pressure by passing through a contracted passage, as in a stop-valve partly closed, or in the common "throttle-valve," it is said to be "throttled," and is shown by the fall of the steam line *k* to *e*, as exhibited in diagrams, Figs. 11 and 16.

The term "wire-drawing" is almost identical in meaning with throttling, but refers especially to the slow cutting-off of steam by an ordinary slide-valve; the result in the diagram being a gradual slanting downwards of the steam line until it passes imperceptibly into the expansion line. Diagram Fig. 22 is an example of this; and the dotted lines show the effect of a quick

cut-off obtained by means of an expansion valve.

With the ordinary valve-gearing, especially the shifting link in common use in locomotive engines, or when a single eccentric connected directly to the valve-rod is

Fig. 22.



used, it is impossible to obtain an early cut-off without a certain amount of wire-drawing. If, under these circumstances, an earlier cut-off than half-stroke is attempted, wire-drawing becomes excessive.

The above diagram, Fig. 22, taken from

one of the most advanced types of locomotives, exhibits considerable wire-drawing. The dotted line shows the pressure that might have been obtained with the same amount of steam more rapidly introduced into the cylinder, indicating a loss from this cause alone of about *ten per cent* of the whole power of the engines.

Modern automatic cut-off valve arrangements are so designed as to avoid wire-drawing with high rates of expansion ; the commonest and simplest being by means of double eccentrics, one of which is operated by the governor so as to give a sufficiently rapid and early cut-off. See diagrams, Figs. 10, 12, 13, and 15, which show a perfectly steady steam line up to point of cut-off, with expansion through the rest of the stroke.

It is an established fact, that "wire-drawing" and "throttling" are accompanied by direct loss, due to the reduction of initial pressure which takes place during the process, and by indirect waste, owing to the increased proportion of work expended in overcoming the back-pressure.

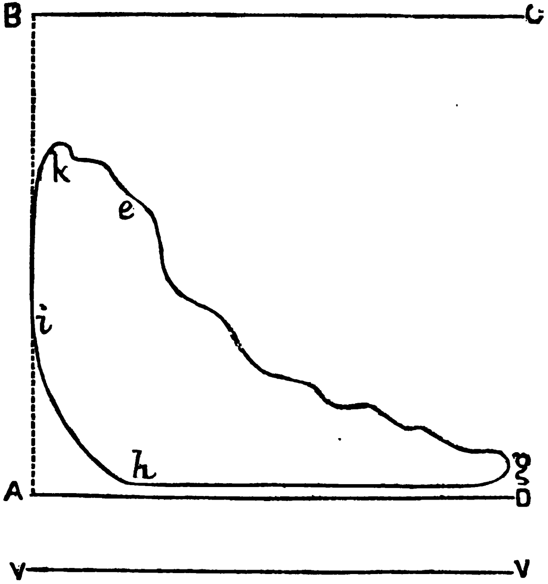
Aside from the economic loss, there is the no less serious objection to contracted passages, that, as the cylinder pressure is reduced (and therefore the power of the engine in the same proportion), a large-size engine becomes only equal to one of less size, weight, and cost, with more liberal steam passages.

UNDULATIONS, OR WAVINESS, OF THE EXPANSION LINE.

The waviness sometimes seen in expansion lines is caused by the inertia of the indicator piston, and in some cases by the use of a weak indicator spring on high-speed engines. (See diagram Fig. 23.) The weaker the spring, the more rapidly the steam will compress it, and consequently the greater will be the velocity of the indicator piston in rising; but the momentum (which is proportional to the square of the velocity) carries the piston above the point to which the steam pressure alone would have compressed the spring. When the momentum has been destroyed by the spring, the spring then forces the indicator

piston below the point where it and the steam would be in equilibrium; and it is again forced too high. These alternate

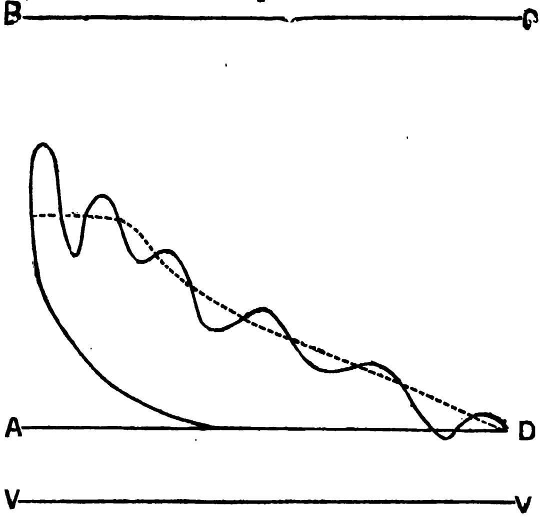
Fig. 23.



up-and-down movements produced by the momentum, combined with the lateral movement of the card, give the wavy line.

These lines are of great value, as they show precisely the degree of suddenness or violence of the action of the indicator.

Fig. 24.

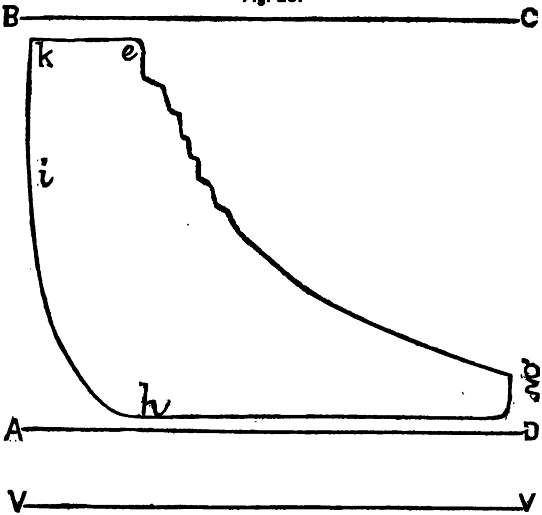


They may occur at the point of admission, of cut-off, and of exhaust.

The reduced diagram, Fig. 23, taken from a high-speed engine running at the Brush Electric-Light Station, Philadelphia,

Penn., in 1882, at 292 revolutions per minute, affords a beautiful illustration of this action.

Fig. 25.



To diminish the extent of these undulations, the spring of the indicator should be stiff, and its mechanism light. These undulations, when excessive, make it extremely difficult to determine the mean effective

pressure from the diagrams when measured by ordinates. To determine the area, it is customary and more accurate to sketch a diagram, freed from these undulations, over the actual diagram taken (as represented by dotted lines in diagram, Fig. 24) *mid-way between the crests and hollows* of the waves. This is better than drawing a line enclosing the same area with the wavy line.

Where the fall of the expansion line is a succession of steps (see diagram Fig. 25), it shows slight friction in the instrument, and that there is no rise of the pencil, — no re-action.

THE EXPANSION CURVE OF INDICATOR DIAGRAMS.

A correct curve does not *necessarily* show an economical engine, since the leakage-out *may* balance the leakage-in, in rare cases, and not affect the diagram. But the opposite is indisputable: that an incorrect curve necessarily and infallibly shows a wasteful engine, to at least the amount calculated upon the diagram.

As indicator diagrams represent the

measure of force, or pressure of the steam in the cylinder at every point of the stroke, the actual card from an engine, as compared with the theoretic diagram (other things being equal), indicates the working value and economy of the engine.

Therefore they should truthfully represent the real performance of the engine. Diagrams vary in form, from various causes; namely, quality or condition of the steam, leakage, condensation, adjustment, and construction, — their influence being most noticeable in the expansion curve. This curve will not, in practice, conform exactly to the true theoretical curve. The terminal pressure will always, under the most favorable conditions, be found relatively too high; the amount being greater as the ratio or grade of expansion increases. Where this is not the case, and the expansion curve of the diagram taken coincides exactly with the theoretic curve, the conclusion cannot be otherwise than that the leakage is greater than the re-evaporation; but, in the present state of the arts, there are no practical means of working steam

expansively, and preserving the exact temperature due to the pressure while expanding.

When the expansion curve falls throughout its entire length below the hyperbolic or theoretical curve, it is evidently due to leakage. The expansion curve of the indicator diagram in all ordinary cases terminates above that of the theoretical curve; in fact, sometimes far above it, due to the re-evaporation of the moisture in the cylinder. An engineer, when indicating an engine, should see to it that the piston and valves are tight. Unless they are so, the diagram will not indicate what the engine is really doing, and the engineer cannot ascertain the causes of any peculiarities in the form of the diagram.

CONCLUSION.

It is hoped that enough has been said to present a general view of the application and use of the indicator, and before closing it may be useful to append a few general remarks.

Rankin, Graham, Nystrom, and Porter, in their books on the steam-engine and the indicator, discuss a large number of causes which influence the form of the indicator diagram.

First, The steam pressure undergoes some fall during the passage from the boiler to the cylinder. The amount of such fall varies greatly in different engines; but the general result is that the highest average indicated steam pressure before expansion begins is some two or three pounds less than the boiler pressure.

The most important points to be noticed are: —

(*a*) The resistance of the steam pipe through which the steam passes.

(*b*) The resistance of the throttle-valve.

(*c*) The resistance due to the ports and steam passages; and here, also, the bends or sharp angles, as well as the imperfect covering, of the steam-pipe must be taken into account.

All authorities agree that in the present state of our knowledge it is impossible to calculate separately the losses of pressure due to these causes; and, if it were possible, the resulting formulæ would be too complicated to be of much use. An observation of this kind has a wide application. It may be pointed out, that steam which has been lowered in pressure by the resistance of passages (or has been *wire-drawn*, as we have termed it) is, to some extent, *superheated* by the friction of its molecules, the tendency of all friction being to produce heat.

Second, There is, in practice, a rounding of the angle at *e* (see diagram Fig. 20), at which the expansion curve begins. This is called *wire-drawing at cut-off*. It is always to be seen where the steam-valve closes gradually, as in diagram Fig. 22; but is

reduced to a minimum in the improved form of cut-off valves as are in general use, as Buckeye, Porter, Allen, and Corliss engines. Speaking generally, it may be said that the steam begins, as it were, to work expansively a little before the valve is completely closed ; or, that the power exerted is nearly the same as if the valve had closed instantaneously at a somewhat earlier point of the stroke, which point may be termed the "effective cut-off." Such a point is easily obtained by carrying the expansion curve a little higher, and by prolonging the probable steam-line to meet it.

Third, The rounding of the expansion curve (see diagrams, Figs. 17 and 20, at *f* to *D*), when release begins before the end of the stroke ; and it is recommended that the point of exhaust release should be so adjusted that one-half of the fall of pressure takes place at the end of the forward stroke, and the other half at the beginning of the return stroke (see *D*, *d*). Where the release is small, the expansion curve is continued to the end of the diagram (see Fig. 18).

Fourth, The general effect of water in the cylinder, from whatever cause produced, but which we will suppose to be present in some degree throughout the stroke, is to lower the steam-line in the first portion of the stroke, and to raise it in the latter portion.

Fifth, There is also the conduction of heat to or from the walls of the cylinder, the general effect of which is that in the last case.

Sixth, Clearance will modify the form of the expansion curve of steam, by removing backwards through a small space the zero line of volumes (see diagram Fig. 17); and, as we have seen, if the steam be completely exhausted from the cylinder during the return stroke, the effect of clearance is to waste a quantity of steam during the double stroke (see diagram Fig. 15). But, inasmuch as it is possible to compress a portion of the exhaust steam in the cylinder during the return stroke (see diagrams, Figs. 18, 20, 22, and 23), the loss above referred to may be greatly or perhaps wholly eliminated.

The best authorities on this subject recommend that the point of compression should be adjusted in such a manner that the quantity of steam confined or cushioned should be just sufficient to fill the clearance spaces with steam at the initial pressure, when the piston comes to rest. In such a case the work expended in compression is restored again during expansion, and the steam spring is continually reproduced without waste.

Seventh, It will be seen by diagrams, Figs. 11, 16, and 22, that *throttling* and *wire-drawing* are accompanied by direct *loss* due to the reduction of the initial pressure which takes place during the process, and by indirect waste owing to the increased proportion of work expended in overcoming the *back-pressure.*

Eighth, There is a great necessity for a delicate steam-engine indicator, giving continuous diagrams on a roll of paper, similar to the stock-quotation indicators.

By

No. 28.—ON TRANSMISSION OF POWER B WIRE ROPE. By ALBERT W. STAH. Fully illustrated.

No. 29.—INJECTORS; THEIR THEORY AND USE. Translated from the French of M. LEON POUCHET. Illustrated.

No. 30.—TERRESTRIAL MAGNETISM AND THE MAGNETISM OF IRON SHIPS. By PROF. FAIRMAN ROGERS. Illustrated.

No. 31.—THE SANITARY CONDITION OF DWELLING HOUSES IN TOWN AND COUNTRY. By GEORGE E. WARING, Jr., Consulting Engineer for Sanitary and Agricultural Works.

No 32.—CABLE MAKING FOR SUSPENSION BRIDGES, as exemplified in the C struction of the East River Bridge. WILHELM HILDENBRAND, C. E. Fu illustrated.

No 33.—MECHANICS OF VENTILATION. I GEORGE W. RAFTER, Civil Engineer.

No. 34.—FOUNDATIONS. By PROF. JULES GAUDARD, C. E. Translated from the French, by L. F. VERNON HARCOURT, M. I. C. E.

No. 35.—THE ANEROID BAROMETER: Its Construction and Use. Compiled by Prof. G. W. PLYMPTON. 2d Revised Edition. Illus.

No. 36.—MATTER AND MOTION. By J. CLERK MAXWELL.

No. 37.—GEOGRAPHICAL SURVEYING: Its Uses, Methods and Results. By FRANK DE YEAUX CARPENTER.

No. 38.—MAXIMUM STRESSES IN FRAMED BRIDGES. By Prof. WM. CAIN. Illustrated.

No. 39 —A HAND-BOOK OF THE ELECTRO-MAGNETIC TELEGRAPH. By A. E. LORING, a Practical Telegrapher. Illustrated.